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TRANSACTIONS

OF THE

INSTITUTION OF NAVAL ARCHITECTS.

TRANSACTIONS

OF THE

INSTITUTION OF NAVAL ARCHITECTS.

VOLUME LV.

PART II.

EDITED BY

R. W. DANA, M.A., M.Inst.C.E.,
SECRETARY OF THE INSTITUTION.

1913.

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INTRODUCTORY PROCEEDINGS.

The Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects were held at Glasgow in the Rankine Hall of the Institution of Engineers and Shipbuilders in Scotland on June 24, 25 and 27, 1913.

The Members were received by the Right Hon. Lord Inverclyde, Chairman of the Reception Committee, and the following address of welcome was delivered by Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland:—

Mr. E. Hall-Brown (President of the Institution of Engineers and Shipbuilders in Scotland): My Lord President and Gentlemen, I have a very pleasant duty to perform on behalf of the Institution of Engineers and Shipbuilders in Scotland in extending to the Institution of Naval Architects a most hearty welcome to these rooms, and I hope that your visit to Glasgow and district and your meeting within these buildings may be both pleasant and profitable.

This is not the first time upon which your Institution has honoured Glasgow with a visit, but it is the first time we have had the pleasure of receiving you within these walls, and I trust that you will find the preparations which have been made for your comfort and convenience such as will facilitate your business and add to the enjoyment of your visit.

In addressing the members of the Institution of Naval Architects I am addressing those upon whose work very much of our prosperity and happiness as a nation depends. Living in our island home, the science and practice of Naval Architecture are of the most vital importance to us. Whether we turn our attention to that branch of your profession which relates more particularly to the defence of our country, our Colonies and our commerce, or to that which deals with the ocean carrying of the world, our Mercantile Marine, we are bound to recognise the primary importance of your work.

As men of peace, we all heartily desire the time when the sword and the spear will be converted into the ploughshare and the pruning hook, and men shall study war no more; but we are forced to acknowledge the inflexible logic which compels you to give much of your labour to the more warlike side of your profession. As Naval Architects, the ethics of the question do not concern you whatever your opinions may be regarding the imminence of the time when the universal brotherhood of man will be recognised and the duties of brotherhood joyfully accepted by all the nations of the earth. What does immediately concern you is that while warships have to be built it is your duty, and one in which you will never, I am sure, be found lacking, to design, build and equip the best that human ingenuity can produce.

Although the problems connected with vessels of war must engage the attention of many of your members, you will, I think, agree that the Mercantile Marine of the world is the most important field for your energies. The vessels which carry the commerce of the world and maintain our connection with the most distant lands are of more real importance than the vessels which are built to defend them, and the problems involved in their construction and propulsion are not less important than those in connection with naval vessels.

The developments which have recently been made in the propulsion of ships have introduced new problems, some of which are only partially solved. The steam turbine has made possible speeds which had never previously been attained with vessels of commerce and war, and still more recently the introduction of the internal combustion engine for marine propulsion has produced new problems for solution. In all the fields open to your energies it is evident that the problems requiring your attention are increasing in difficulty day by day, and the skill and ingenuity required from our naval architects and marine engineers are of a higher and ever higher order.

It is in respect to the mental equipment for your work that such meetings as you are now proposing to hold find their complete justification. Whether we consider the papers which will be read and discussed, the visits which will be made to the various engineering and shipbuilding establishments, or to the more social events in which you will participate, from each and all you will return better equipped and more mentally fitted for the labours of your profession than before.

The Institution of the Naval Architects, like the profession of Naval Architecture, is an international one, and it is a great pleasure to note that among our visitors we have the privilege of including representatives from other lands. The profession of Naval Architecture owes much to the labours of our talented confrères in Italy, France, and Germany, and in still earlier times to those of Portugal, Spain, and Holland. Indeed, it would be impossible to name any maritime nation which has not contributed to the progress of Naval Architecture in either ancient or modern times. The sea is indeed the highway of the nations, and interest in the progress of Naval Architecture world-wide. It is, therefore, a special pleasure to welcome those who have come from other shores to participate in the business of your meetings and to honour us with their presence.

The arrangements which have been made for your Summer Meeting are a happy blending of business and pleasure. We all recognise that "all work and no play makes Jack a dull boy," and as dull boys are of absolutely no use, at least as Naval Architects, I trust your meetings will result in a brighter, as well as a keener, mental outlook on the part of everyone taking part in them.

I have endeavoured to express to you the pleasure it is to me to welcome you to these rooms in the name of my Council and the Institution as a body. I can assure you that the proposal that your Institution should hold its Summer Meetings in Glasgow as our guests at once met with the heartiest approval of my Council, and I trust that the success of the meeting may be as great as the heartiness of the welcome which in their name I now extend to you.

The PRESIDENT of the Institution of Naval Architects then took the chair and delivered the following reply :—

The Most Hon. the Marquis of BRISTOL, M.V.O. (President) : Mr. President, my Lords, Ladies and Gentlemen, it has been a great pleasure to me to listen to the words of welcome that have fallen from the President of the Institution of Engineers and Shipbuilders in Scotland. When the Council of that Institution signified its readiness to welcome our members in Glasgow, and to open the doors of its handsome building for our Summer Meeting, it was felt that the success of this meeting was assured. That feeling has been strengthened by a perusal of the programme that has been arranged,

which contains the promise of a most instructive and agreeable series of meetings, visits and entertainments. As to the latter, it was certain to all who have experienced the wealth of Scottish hospitality —and who of our members has not, at one time or another ?—that the social side of these summer meetings would be an outstanding feature of the programme, and this expectation has been fully justified. Our warmest thanks are due to the Chairman and members of the Reception Committee, and to all those who have by their voluntary efforts contributed to the success of these meetings. But the social side of such gatherings, however pleasant, does not constitute their principal or most valuable feature. Although the reading of papers may be considered by some to be in the nature of a necessary evil on such occasions, yet it must be remembered that they constitute valuable contributions to the permanent records of the work of the Institution. I think it will be admitted that in this respect the list of papers before us promises to add in no small measure to the value of this year's Transactions. Apart from this feature, however, summer meetings provide in themselves a very agreeable means of bringing together those who, united by bonds of common interest, may, by the nature of their calling, be precluded from meeting except at rare intervals. This is particularly the case with those of our members who reside abroad, to whatever nationality they belong, and to these we extend a hearty welcome whenever they are able to join us, whether in Glasgow, London or elsewhere.

In the present instance, however, our summer meetings also provide a common ground for the members of our respective Institutions, whose origin dates from so nearly the same period, and whose aims and objects are in so many respects alike. The cordial relations which have always subsisted between these two Societies, and the flourishing condition of both, are proofs that the field of work in which they are engaged is a large one, large enough for many to take part in its development without overlapping or clashing of interests. The friendly rivalry which is maintained by the existence of such separate corporate bodies is in itself an incentive to better progress and freer development than would obtain under any system of amalgamation or affiliation, while the presence of so many members common to both Societies ensures a harmonious working and mutual co-operation towards the attainment of a common goal. Both Institutions have furnished distinguished representatives upon the various public and private committees that have done, and are doing, so much to promote the advancement of naval architecture in its various branches, and to obtain recognition of its aims and aspirations. It is especially pleasing in this connection to note that the high distinctions which have so recently been conferred by His Majesty the King upon the Chairmen of two of these important Government Committees have fallen to two of the most prominent and capable members of both our Institutions. In Sir Archibald Denny, Bart., no happier selection could have been made than that which will perpetuate in the Denny family the highest traditions of an industry which has been developed through succeeding generations by a blend of science and practice for which the members of that distinguished family have long been famous. The knighthood conferred upon Sir John Biles is a recognition of his numerous public services on Government and other committees, and of his many-sided activities in shipbuilding technology. Both these distinctions serve to enhance the prestige of naval architecture, and shipbuilders and marine engineers cannot but feel that their labours are stimulated by such recognition of their value.

It is now twelve years since our Institution held its last Summer Meeting in Glasgow, and those years have indeed been memorable ones in the shipbuilding world. Time does not allow me, even if your patience did, to refer to the many remarkable events that have occurred in the interval: events that have turned the current of our thoughts into new channels, and developments that have left their mark upon the vessels that bear the burden of armaments, as well as upon those hardly less costly and even weightier ones that carry to and fro the world's commerce upon the face of the

waters; changes in design and methods of propulsion that have altered the policy of nations and remodelled the strategical requirements of our fleets, that have called for the greatest skill and courage on the part of designers and builders, and which have justified by their successful application and their general adoption by others the care and forethought bestowed upon their inception.

Now the Clyde, the acknowledged birthplace of steam navigation, has long maintained her pride of place as the greatest of our shipbuilding rivers—a stream whose banks have cradled some of the finest examples of the shipbuilder's art, both in the days when it was merely an art, and now in these more strenuous times when science has superseded art as the foundation of naval architecture. To those who, like myself, come only at intervals to this great centre of shipbuilding activity, it is an inspiring sight to witness in their many stages of construction the variety of craft that may be seen here. No one, I think, can gaze unmoved at the teeming cells in this hive of industry, or watch without a thrill of pride the birth and growth of these future messengers of England's sea-borne power and wealth. That many of these developments in ship construction can be ascribed to the labours of the members of our Institutions is a source of legitimate pride and satisfaction. How far we may be able to trace their inception to any single paper read within these or similar walls it is, of course, impossible to say, but it is certain that much healthy stimulus has in the past been given by full and frank discussion of these engrossing topics. Two instances of the direct practical outcome of such discussions stand out in connection with the last occasion when we met in Glasgow. Those of you who were then present will no doubt remember that two resolutions were put to that meeting. The first was the result of a paper by Lord Brassey on "Mercantile Auxiliaries,"* and was proposed by the late Admiral Sir John Dalrymple Hay to the effect that, "As our supremacy depends on the efficiency of our Naval and Mercantile Marine, a committee of Admiralty officials, shipowners, and shipbuilders should be formed to discuss the best method of constructing a combined naval and mercantile marine." This resolution was followed up by representations made by the Council of the Institution to the Admiralty, which eventually led to the formation of the Government Committee on Mercantile Cruisers. The Institution was represented on this committee by Professor (now Sir John) Biles, and the committee's report led the Government of the day to take measures which provided for the subsidising of the Cunard Company, for the purpose of building the *Lusitania* and *Mauretania*.

These facts are no doubt familiar to many of you, but the effect of these measures has been far reaching, for there has sprung up a better and closer understanding between our naval and merchant services, arising out of a fuller recognition of the value of our great steamship lines to naval needs, both as auxiliaries in time of war and as a splendid training ground for naval reservists in time of peace. Thus the problem of safeguarding our merchant ships in war time has been more hopefully attacked since measures have been taken for helping them to help themselves, and the country reaps the advantage of a broader field to draw upon for manning its ships in time of war.

The second resolution, which was passed at that same meeting of the Institution in this city in 1901, was proposed by Mr. A. F. Yarrow, for the establishment of a national experimental tank, and it eventually led to his generously endowing the National Physical Laboratory with the splendidly equipped tank with which you are all familiar. The value of the work which is being done there is now obtaining general recognition, and, apart from the extremely interesting papers which have been contributed from time to time to our Transactions on the researches carried out there, the tank is now in frequent demand for work arising from inquiries of private shipbuilders, who are deriving advantage from being able to get models made and tested there.

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* Trans. I.N.A., Vol. XLIII., page 223.

Results such as these, Gentlemen, surely afford sufficient justification, if any were needed, for the holding of our summer meetings. My hope is that the meetings which we are now inaugurating may prove as fruitful in good results for the advancement of naval architecture as were those of twelve years ago. We have, alas! since that time lost some of the best and brightest intellects that adorned our roll of membership, but our ranks keep filling up as fast as the gaps occur, and we look to the younger generation to maintain the noble traditions of the past. No district of Britain has done more than this part of Scotland to supply men of the right stamp, men full of grit and enthusiasm, and of confidence in the future, and we look to them to carry on with success the great works of construction that have rendered the Clyde famous in the annals of British shipbuilding—not merely shipbuilders, but Empire builders as well!

Presentation of a Gold Medal to the Institution of Naval Architects in Recognition of Services Rendered to the Advancement of Naval Architecture.

Signor Salvatore Orlando (President of the Italian Society of Naval Architects): My Lord, Ladies and Gentlemen, the Collegio degli Ingegneri Navali e Meccanici Italiani, which I have the honour to represent (together with my friends, Colonel Russo, Captain Barbé, and Signor Giacomuzzi) at their general meeting of members, held in Spezia on December 22 last, resolved to present the Institution of Naval Architects with a gold medal, expressly struck for the purpose. This medal is, no doubt, a very modest gift when compared with the great value of the Institution in furthering the progress of naval science, but we hope nevertheless that it will be accepted as a token of our sincere friendship and goodwill. The Collegio desires also to thank the Institution for the valuable and highly-appreciated gift of its past volumes of Transactions, and also for the very kind hospitality which has always been extended to Italian Naval Architects and Marine Engineers at these meetings. We hope in the near future to see British Naval Architects attending our meetings in Rome. Italy to-day is not only the land of an ancient civilisation, but she has become an industrial state in which steel, armour, guns, engines, and ships are made and built in Italian workshops, so that English engineers travelling Southwards to enjoy our sunshine, may, I hope, also be interested in my country from a technical point of view. If such an invitation is acceptable, we will, by all means in our power, do our best to welcome you most heartily, for we feel deeply indebted to the Institution of Naval Architects for the cordial welcome which we have always received at the hands of its members.

The President: Ladies and Gentlemen, I am sure you will wish me at once to thank Signor Orlando for the trouble which he and his confrères have been put to in travelling such a great distance to present this medal to the Institution of Naval Architects. I wish that I had Signor Orlando's gift of tongues so as to reply in his own language; but as I am unfortunately incapable of doing so, such few words as I propose to say I shall put in plain Anglo-Saxon.

First, as I say, I wish to thank Signor Orlando and Captain Barbé, of the Italian Embassy in London, and Colonel Russo, for coming here to-day to present this beautiful medal. I see, on looking at it, that it brings up to date the science of naval architecture and marine engineering. Not only are

pictures of the most modern ships and turbines engraved upon this medal, but they are most artistically reproduced, as we might expect would be the case with anything coming from Italy, that great nation which has always been to the forefront in Naval Architecture. When I was a young naval officer I was given a lecture to deliver on the difference between three different classes of ships; one was an English ship of the *Citadel* type, another was one that had a complete belt of armour, and the third was one of those great ships of the Italian Navy, the *Italia* or *Lepanto*, which formed a special type at that time, and whose origin was then due entirely to the versatility of Italian genius. Their armament included huge 100-ton guns, a fact which made us go one better and try and secure our predominance by building 111-ton guns. May I now express the hope that this Italian Society of Naval Architects —which I believe is still quite a young society but a very vigorous one, and which is increasing in numbers every day—may we hope that if they do invent any new fearsome engines of destruction they will keep them in a secret corner of Pandora's box, safely locked up, so that we shall have no further necessity to compete against them by making still more effective engines of destruction. Let them open wide that other side of Pandora's box, and give to the world all the advantages we look and hope for from any progress of science that improves our mercantile marines, fosters our trade and commerce, and does good to the whole world. Let us all try and make for the comfort and well-being of all, whether they are in the Royal Navies of the different countries or in their mercantile marines; let us make for peace and contentment rather than look for new methods of destruction. Gentlemen, I thank Signor Orlando very heartily for this very beautiful gold medal, which I should be proud to wear if the rules of the Institution of Naval Architects allowed me to do so!

NEW MEMBERS.

The following gentlemen, having been duly recommended, were unanimously elected :—

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Elliot, John, Managing Partner of Messrs. Elliot & Jeffery, *Cardiff.*
Elliot, Thomas, Managing Partner of Messrs. Elliot & Jeffery, *Cardiff.*
Harvey, Bertram, Engineer Lieutenant R.N., Admiralty, *London.*
Humby, Henry George, Superintendent and Consulting Engineer to the Union of South Africa Railways, *Victoria Street, London, S.W.*
Iwasa, Thoichi, Commander Constructor, Chief Inspector of Hull Department for I.J.N. at Messrs. Yarrow & Co., *Glasgow.*
Laurenti, Cesare, Technical Director of the Fiat Shipyard, *San Giorgio, Spezia.*
Morgan, Charlton Elliot, Assistant Manager, Messrs. Vickers, Ltd., *Barrow-in-Furness.*
Murdoch, Richard Alexander, Superintendent to the British India Steam Navigation Co., Ltd., *Bombay.*
North, Dudley Henry, Constructor, Royal Indian Marine, *Bombay.*
Rodger, James Allison, Marine Engineer to the Great Central Railway Co., *Grimsby.*
Strachan, Robert, Assistant Chief Draughtsman to the Fairfield Shipbuilding Co., *Glasgow.*
Takesaki, Tomokichi, Engineer Commander, Chief of the Machinery Inspection Department, I.J.N., at Messrs. Yarrow & Co., *Glasgow.*
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Associate-Members.

†Brisbane, John Simpson, Head of Scientific and Design Staff to Messrs. W. Denny & Bros., *Dumbarton.*
Chapman, Laurence Boylston, Designer with the Holland Torpedo Boat Co., *New London, U.S.A.*
‡Child, Geoffrey, Assistant Constructor, Admiralty, *London.*
Dight, Sydney Rupert, Engineer Lieutenant, R.N., Admiralty, *London.*
Eidlitz, Cornel, Assistant Superintendent Engineer with the Royal Hungarian Sea Navigation Co., *Fiume, Austria.*
Fenton, Frank, Chief Draughtsman to the Fore River Shipbuilding Co., *Quincy, Mass., U.S.A.*
†Flockhart, John Henry, Manager for the Manchester Dry Dock Co., *Cheshire.*
Hunter, Edward, Assistant Manager, Osaka Iron Works, *Osaka, Japan.*

* Transferred from Associate-Member Class.
† Transferred from Associate Class. ‡ Transferred from Student Class.

McLaren, William Dick, Engineering Department of the Fairfield Shipbuilding Co., *Glasgow.*
Matheson, Stanley, of the Hudson Bay Steamship Co., *Montreal.*
Meek, Arthur Rowland, Partner in the firm of Messrs. Nicholas & Meek, Consulting Engineers, *Hull.*
Mörch, Hans Ramm, Professor of Naval Architecture at the Technical High School, *Trondhjem, Norway.*
Murray, Athole James, of the New York Shipbuilding Co., *Camden, U.S.A.*
Paffett, William Frederick, Engineer Lieutenant, R.N., Admiralty, *London.*
Robson, William John, Superintendent Engineer to Messrs. Elvidge & Morgan, *Cardiff.*
Rolland, John, Surveyor to Lloyd's Register, *Vienna.*
Rutland, George Henry, Draughtsman, Messrs. Yarrow & Co., *Glasgow.*
‡Stark, Alexander, Jun., Ship Draughtsman with the Fairfield Shipbuilding Co., *Glasgow.*
Welsh, George Muir, late of the Fairfield Shipbuilding and Engineering Co., *Glasgow.*
Young, Wilfred Edward, Assistant to Works Manager, Messrs. Yarrow & Co., *Glasgow.*

Associates.

Abo, Captain Baron Kiyokazu, I.J.N., Naval Attaché to the Japanese Embassy.
Bainbridge, Captain Robert S.
Bentinck, Captain Rudolf W., R.N.
Browning, Rear-Admiral Montague E., M.V.O.
Colmore, Reginald B. B.
Dick, William Henry
Giacomuzzi, G.
Gillmor, Reginald Everett
Kincaid, James Scott
Lilliehook, C. J. F. M., Professor
Lund, Frederic William
Minett, Harry Frederick
Phillips, Stephen
Somers, Frank
Sumner, Percy James Hammond

Students.

Holzapfel, George Leopold
Hooton, Henry Hurst
Kidd, Eric Leslie
Neilson, Archie McNicol
Reed, John A.
Shen, Chen Tung
Wilson-Jones, Douglas

‡ Transferred from Student Class.

ON SHIPBUILDING CONTRACTS.

By L. PESKETT, Esq., Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 24, 1913; Rear-Admiral the Most Hon. the Marquis of BRISTOL, M.V.O., President, in the Chair.]

IT was the original intention of the author to deal in this paper with the question of "Ship design from the Shipowner's point of view," and this title was given in the official programme of this Meeting. As the paper developed, it was found desirable to limit its scope, and to reserve for some future occasion the consideration of a number of technical matters which were being included.

The first subject dealt with under the original title was "The Making of Contracts," and the author has decided to confine himself to this section on the present occasion. The object of this part of the paper is to lay open for discussion the method usually adopted in making shipbuilding contracts.

To members of the Institution of Naval Architects, the relationship that exists between the shipowner and the shipbuilder is one of peculiar interest, as the enterprise of the owner has considerable influence on the activity of the builder, the function of the naval architect being that of technical ambassador between the two. In this capacity he carries a burden of anxiety, for it is his duty to attain the realisation of those utopian ideals which the owner invariably associates with a characteristic dread of expenditure, by obtaining from the builder a ship complete in all its details, and of full value for the amount expended.

It needs but little thought to appreciate that it is in the best interests of our country to keep the cost of production within reasonable limits. When, at the launch of the *Aquitania*, the chairman of the builders' company asked for support in the formation of a combination among shipbuilders, he was appealing on behalf of a system which is the antithesis of healthy trade. The ultimate effect of any such combination is to increase the cost of production, which would be disastrous to the shipbuilding industry. The shipbuilding markets of the world are gradually altering their attitude with regard to Great Britain and countries that at one

time relied upon her to build their ships are now successfully competing with her in obtaining orders, even from British owners. The maintenance of both our shipowning and shipbuilding supremacies depends on the fundamental principle that the owner must only pay that price to the builder which is sufficient to assure him a fair and reasonable return. This is essential if the shipbuilding industry is to remain our inheritance, for every increase in the cost of production assists others in wresting it from us.

It will be well when those who control the various trade unions come to realise this important fact. Labour troubles have of late years been a source of great anxiety to shipbuilders. The time devoted by the heads of firms to the settlement of petty disputes has meant a reduction of efficiency in the conduct of their business. A solution of these difficulties might be found in the establishment of a disinterested tribunal, whose judgment in all cases should be agreed to mutually. The time at present spent in settling disputes could be more profitably employed by imitating the procedure of owners and their superintendents, who periodically hold international meetings to discuss details in connection with the work in all departments of their business.

The object of the owner when negotiating for the construction of a ship is to obtain the best revenue-producing asset it is possible to get for the sum he intends to invest. The leading shipowning companies prepare their own plans and specifications, upon which builders are invited to tender. The tenders thus received are comparable, being all prepared on a common basis.

The contract, as usually drawn up, specifies a certain fixed sum to be paid for the construction of a ship of given dimensions and speed, to be built to the requirements of the owner, and in accordance with the regulations of a particular registration society. Compare this with the procedure followed when contracting for a building.

When a building is to be erected, plans and specifications are prepared by an architect. When these have been approved, they are passed on to a quantity surveyor, who prepares a bill of quantities. Prices are then obtained for the supply and erection of the various items, based on some standard of measurement. When the building is finished the actual amount of work done is measured by the surveyor, and payment made on this assessment to each contractor at the rate previously agreed upon. By this method the owner of the building may during the progress of the construction decide to add to, or take from, any of the scheduled items, with the knowledge that there will be a corresponding addition or reduction in the final cost.

There appears to be no reason why such methods should not be applied to shipbuilding. At the present time every firm has its own particular method of preparing estimates, but it should not be a difficult matter to arrange a suitable standardised form

for computing the cost. If such a plan were adopted, the various tenders could be submitted on a standard basis. For example, all the items in connection with the cost of material and labour, establishment charges, profit and margin, could be scheduled, and a total sum defined as the maximum price which the owner would be required to pay. If the margin on direct outlay, the establishment charges and the net profit were agreed upon as a fixed sum, it would be an advantage to both owner and builder to keep the cost as low as possible. For since these items do not alter, the builder makes the same profit on a smaller turnover, while any reduction in the net cost of material and labour from the amount originally fixed reduces the cost of the vessel to the owner.

Such a system has been already carried out in practice by the Cunard Company in the case of the building of the *Franconia*. When it was arranged for that vessel to be built by Messrs. Swan, Hunter & Wigham Richardson, Limited, a contract was made in the usual way, that is, a fixed sum was agreed upon to complete the ship in a given time. When the contract was signed, it was suggested that the ship might be built on the principle outlined above, the price already quoted being considered to be the maximum sum payable by the owner. This course was agreed to, and the result of the experiment was that upon the completion of the contract the builders handed to the owners a rebate of £20,000 on the sum originally fixed. This gave satisfaction to all concerned, although to the uninitiated it might be imagined that it meant a loss to the builder. The reverse was actually the case, for the original profit named remained fixed and the rebate was accounted for in the savings on material and labour mutually agreed to during the construction of the ship.

In Table I. (facing page 8) an attempt has been made to classify under convenient headings the total cost of building a ship. It will be noticed that the Contract Price has been divided into four general sections, namely :—

(1) Direct Outlay.
(2) Establishment Charges.
(3) Scheduled Items.
(4) Builder's Profit.

The Contract Price would generally include all the work under these four headings, although the decoration of the public rooms included in Section 3 might be made the subject of a separate contract, and entrusted to a firm of decorators and carried out in accordance with the designs of an architect. In addition to the Contract Price, the incidental establishment charges incurred by the owner's staff in the performance of work connected solely with the building of the ship, such as salaries, travelling, office, and other expenses, as well as the cost of any items which the owners might themselves prefer to supply in order to complete either the equipment or outfit, would

have to be added, in order that the whole estimated cost of the undertaking could be ascertained.

Taking these sections in order, it will be noted that the "Direct Outlay" is subdivided under three headings, namely, hull and outfit, machinery, and sundry fees and expenses. The cost of the hull and outfit is again subdivided under headings for materials, wages, and equipment and outfit. The cost of machinery is treated similarly, engines and boilers being each separately classified under the headings of material and labour: in the case of the engines the main and auxiliary portions are kept as separate items.

Under the heading of "Establishment Charges" have been classified various salaries and facilities, together with sundry items of gear which the builder is bound to provide in order to keep his establishment open, and for the use of a proportion of which payment is due to him from his customers for the time being.

The "Scheduled Items" which form the next subdivision are those usually associated with sub-contracts which may, or may not, be carried out by subsidiary departments of the builder's establishment. The interpretation of the meaning of these items is generally a matter of considerable discussion, after the signing of a contract. It is in every way preferable for the owner to indicate, and to have allotted by the builder, a prime cost amount for each item tabulated. The cost named should be included in the builder's estimate, but the owners would ultimately pay an extra, or receive a rebate, according as the net cost to the builder is more or less than the sum originally allocated.

The inclusion, or otherwise, of the cost of the "Decoration of the Public Rooms" is a matter which would be determined by the nature of the work involved. In the case of the very elaborate schemes associated with large liners, this work is undoubtedly better placed in the hands of some firm of sub-contractors.

The succeeding Tables, Nos. II. to V. (page 8), serve to illustrate a manner in which the "Contract Price" might be determined in detail. The first sum which should be computed is the estimated cost of the builder's "Direct Outlay."

Dealing first with the cost of material, Table II. outlines a method whereby the cost of the steel work and such material can be determined, which is generally done on the basis of weight. Shell and deck plating would be dealt with first, and then such other plating as occurs throughout the structure. Thereafter, and in a systematic manner, the quantities of angle sections would be taken out. An estimate would be made for the weight of various classes of forgings and castings, based on previous experience with similar ships, and this, with all other quantities, would be subsequently revised when the items came to be weighed.

Table III. deals with carpenter's wood, which will for the most part be taken out on a basis of superficial feet.

In the two foregoing tables the cost of material is kept separate from the cost of wages. In some cases, as, for example, in the manufacture and erection of cabins and cabin framing, furniture, painting, and other work, it is a convenient and easy matter to lay down rates for combined material and labour. The method suggested in Table IV. indicates such a treatment. If furniture were to be included the owner could name a price for the various items, and samples could be subsequently discussed and the price amended to suit.

As another example, Table V. illustrates briefly the method of determining the cost of plumber's material.

All the foregoing tables are intended to serve merely as indications of method, and should not be taken to represent a finished scheme. In the limits of such a paper as this no other treatment is possible, and an enormous amount of detail must necessarily remain undiscussed.

In some such perfectly feasible manner the whole cost of material alluded to under the heading "Direct Outlay" could be estimated, and a joint agreement come to with regard to the net sum involved.

The amount due for wages would then be taken out on the basis customary in the district. This can only be done after the amount of material has been determined. In such cases as those alluded to in Table IV. no sum would be involved under this head, as the costs are derived on a basis which includes wages.

The wages included under "Direct Outlay" are those of workmen directly employed in the construction of the ship, or working on machines while these are being used solely in this connection.

To an estimate such as this, it will be generally conceded that a positive margin of, say, 5 per cent. should be added, after a careful computation of the actual amount has been agreed to, in order to allow for errors and omissions. This margin should be added in when defining the cost of the "Direct Outlay."

A sum having been agreed upon as representing the Estimated Direct Outlay upon the ship, the Establishment Charges can be settled. These can be regarded as dependent upon the Direct Outlay, and it will be reasonable to assign for them a sum which is a certain agreed percentage of this amount. What percentage is actually to be taken must depend on the nature and amount of work to be performed, for it is easy to see that establishment charges may be greatly increased if any special facilities, such as additional cranes, piling and dredging are required to carry out the work.

It is now possible to sum the three following quantities, namely :—

(*a*) The estimated direct outlay ;

(*b*) The establishment charges (a percentage of the direct outlay) ; and

(*c*) The sum assigned by the owners to cover the cost of certain scheduled items, which sum the builder has to include in his estimate.

The builder's profit is defined as some agreed percentage of the sum of the three items above. The exact percentage having been agreed upon, the contract price is fully determined.

The progress of the transaction should be watched by an auditor, whose function should be to ascertain and certify the amount actually expended by the builders, and to determine the interest; as well as by a quantity surveyor, who would become responsible to builder and owner for the measurement of quantities.

The actual sum to be paid to the builders, as opposed to the Contract Price, will be the sum of the four following quantities :—

(*A*) The actual certified amount of the direct outlay, provided always that this amount is not greater than the estimated sum (*a*) ;

(*B*) The actual sum named above to cover establishment charges, namely (*b*) ;

(*C*) The actual certified cost of the scheduled items ; and

(*D*) The actual sum named as the builder's profit,

and of these (*B*) and (*D*) will remain as originally arranged. The cost under heading (*A*) will be revised in agreement with the investigations of the auditor and the quantity surveyor, and if, after investigation, the sum of the costs under this heading (*A*) is found to be less than the amount of the original estimate, the balance would be returned as a rebate by the builder to the owner.

Section (*C*) must be revised, and should the cost under this section exceed or be less than the scheduled amount, an adjustment would be made.

With reference to the method of payment, an agreement is generally entered into between the owners and the builder, in which the former undertakes to pay to the latter certain definite percentages of the contract price upon completion of various stages in the ship's construction, a certain final instalment being reserved until some period after delivery.

In the case of a big undertaking where the builder would be called upon to make exceptional outlay before the instalments reached him, and where the owner might have difficulty in laying his hands upon the necessary amount when the time for payment

came, it might be desirable that an arrangement should be entered into, whereby the owner could deposit with the builder a certain proportion of the money which would be due for payment in connection with the forthcoming instalment.

It is apparent that such a mode of contracting as is suggested in the foregoing remarks will stimulate the business capacity of the builder. The amount of profit and the amount paid to cover establishment charges is assured to him, and, consequently, it is to his interest to exert himself to keep down the actual charges. His risks are mainly taken when he is agreeing to contract rates for wages, but it is in his best interests, as well as in the interest of the country, that the cost of production should be kept as low as is reasonable if contracts are not to be placed elsewhere, and, consequently, a scheme which enforces him to constant activity in this respect is fundamentally sound.

The cost of meeting the requirements of Post-Contract Legislation is obviously the subject of an extra-contract agreement between owner and builder.

Both of the above are risks which have not embarrassed the successful working of such a system as has been outlined in the case of the building trades, or on those occasions when it has been applied successfully to the building of ships within the author's experience.

TABLE II.—Steel Work.

Method of Taking out Quantity.	Items Specifying fully the nature of Material, its position in ship, &c., &c.	Rate.		Cost.		
		Material.	Wages.	Material.	Wages.	Total.
By Weight	Steel plates, angles, rivets, castings, forgings, pillars, rails, expanded metals, &c., &c.					
By Catalogue Prices	Bulkhead doors, manhole doors, &c., &c.					

TABLE III.—Carpenters' Wood.

Method of Taking out Quantity.	Material Specifying fully class of timber, size, position in ship, &c., &c.	Rate.		Cost.		
		Material.	Wages.	Material.	Wages.	Total.
By Lineal Feet	Rails, sparring, &c.					
By Super. Feet	Decks, ceiling, hatch covers, gratings, wood bulkheads, grain boards, &c.					

TABLE IV.—Joiners' Work.

Method of Taking out Quantity.	Items. Fully specified, with position in ship and description of all work to be performed on same, including painting and erecting.	Rate, Material, and Wages.	Cost, Material, and Wages.
By Lineal Feet	Cants, head runners, architraves, cornices, mouldings, rails, &c., &c.		
By Super. Feet	Framing, ceilings, &c.		
By Number off	Window boxes, beam casings, pillars, pilasters, doors, ladders, &c., &c.		
	Furniture in detail :— Sofas, toilet tables, night toilets, &c., &c.		

TABLE V.—Plumbers' Work.

Method of Taking out Quantity.	Items Fully specified, with position or use in ship.	Rate.		Cost.		
		Material.	Wages.	Material	Wages.	Total.
By Lineal Feet	Piping (all classes)					
By Weight	Couplings, bulkhead connections, cocks, scupper heads, rough castings, &c., &c.					
By Number off	Bends, traps, &c.					
By Catalogue Prices	Various patent articles ...					

[*To face page* 8.

OWNERS' INCIDENTAL EXPENSES.

CERTAIN CONSUMABLE STORES

Excluded from the specification, which by reason of their being of special design, or their being stock articles, the Owner may prefer to supply himself. Should he desire these articles to be supplied by the Builder, he would have a prime cost value assigned to each, and include them under the other scheduled items.

EXAMPLES.

Blankets
Bolsters
Charts
Chronometers
Crystal
Cutlery
Counterpanes
Earthenware
Flags
Linen
Mattresses
Napkins
Plated Ware
Sheets
Silver, Towels, and any similar items.

SALARIES AND TRAVELLING EXPENSES

Incurred by the owner's staff of technical officers, inspectors, and others in connection with the design and construction of the ship.

SHIPBUILDING CONTRACTS.—TABLE I.

[To face page 8.

TOTAL ESTIMATED COST OF VENTURE.

CONTRACT PRICE.

DIRECT OUTLAY.

HULL AND OUTFIT.

Other than Items for which a prime cost is scheduled.

MATERIAL.

Steel and Iron :—
Forgings
Castings
Plates
Angles and Sections
Beams
Rivets
Smithy Iron
Brass Castings

Wood :—
Carpenters'
Joiners'
Pattern
Template
Mould Loft

Sundries :—
Anti-fouling
Bitumen
Bulkhead doors
Cement and Sand
Galvanising
Glass
Packing
Manhole doors
Paints and Oils
Plumbers' material
Polishers' material
Store goods

Material destroyed in connection with the Building and Launching of the Ship.
Coal and Coke used on board by Riveters and others; also in Boilers.
Power used solely in connection with the Building of the Ship.
Water used on board.
Materials used in constructing the Model of the Ship.

WAGES.

Angle Smiths
Caulkers
Cementers
Carpenters
Drillers and Holeborers
Draughtsmen
Engineers
Electricians
Hammermen
Helpers
Joiners
Labourers
Loftsmen
Liners off
Mould makers
Painters
Pattern makers
Platers
Plumbers
Polishers
Red leaders
Riggers
Riveters
Sheet iron workers
Sawmill workers
Smiths
Sparmakers
Tracers

Men working on Machines while being used solely in connection with the construction of the Ship.
Men employed in erecting and taking down staging directly employed in Launching, and in the preparation of the Ship's Berth.
Foremen, Under Foremen, Timekeepers, Storekeepers and the like while employed solely on the construction of the Ship.
Model makers.
Men employed in lifting and fitting up on board of all the Scheduled Items.

EQUIPMENT AND OUTFIT.

Accommodation ladders
Awnings, Sails, and Canvas
Blocks
Deck machinery connection
Derricks, steel
Hoses and Couplings
Lightning Conductors
Netting and Expanded Metal
Name Plates
Pumps
Rigging
Ropes
Signal Lamps
Speaking Tubes
Sundries

MACHINERY.

ENGINES.

MAIN.

MATERIAL.

Iron Castings
Steel Castings
Brass Castings, rough
Brass Castings, finished
Condenser Structure
Condenser Tubes
Condenser Tube Plates
Condenser Sundries
Pump Rods
White Metal
Smithy Forgings
Pipes (all classes)
*Blading Material
*Silver Solder
*Drums
*Wheels
*Spindles
Shafting
Propellers
Plates and Angles
Pattern Wood
Tanks
Store Goods

* In the case of a Turbine Job.

LABOUR.

Boilermakers
Brass Finishers
Draughtsmen
Fitters
Labourers
Machine men
Painters
Pattern makers

AUXILIARY.

MATERIAL.

Circulating Pumps
Air Pumps
Feed Pumps
Feed Heaters
Hotwell Pumps
Fire and Bilge Pumps
Ballast Pumps
Sanitary Pumps
Sundry Pumps
Filters
Aux. Condenser
Fans and Motors
Evaporators
Spare Gear
Ash Hoists
Ash Expellers

LABOUR.

Boilermakers
Brass Finishers
Fitters
Laborers
Machine men
Painters
Pattern makers

BOILERS.

MATERIAL.

Furnaces
Plates
Stay Tubes
Plain Tubes
Stay Bars
Castings
Manhole Doors
Rivets
Fire Bars, &c.
Mountings
Covering
Forced Draught Fittings
Smoke boxes
Uptakes

LABOUR.

Boilermakers
Brass Finishers
Fitters
Labourers
Machine men
Painters
Pattern makers

FEES AND EXPENSES.

Arbiter's Fees
Board of Trade Fees
Classification Fees
Delivery
Docking Dues
Dredging
Insurance
Launching
Lightering
River and Harbour Dues and Fees
Royalties on all Patents
Testing
Towing
Travelling Expenses
Trial Trip Expenses

ESTABLISHMENT CHARGES.

SALARIES AND WAGES.

Officials
Clerks
Works :
Enginemen
Firemen
Foremen
Furnacemen
Gatemen
Hammershafters
Ironworkers
Labourers
Machinemen
Messengers
Photographers
Police
Stocktakers
Storekeepers
Timekeepers
W.C. Attendants
Watchmen
Waggon Weighers
Weighmen
Fire Brigade
Cooks
Stewards

FACILITIES.

Use, Maintenance and Depreciation of—
Berth
Buildings
Dock
Gear
Launching Ways
Machinery
Plant
Advertising
Carriage and General Haulage
Cleaning
Charities
Drawing Office Materials
Fire Appliances
Heating
Insurances for Establishment
Lighting
Photographic Materials
Power
Purveying
Rents
Rates
Stationery
Taxes
Telephones
Telegrams

SUNDRY MATERIALS.

Candles
Cleaning Gear
Machine Shop Consumables
Oils, Tallow, Grease
Water
Waste

SCHEDULED ITEMS,

Concerning which the owner should indicate a Prime Cost value to be included by the Builder in his Tender. The Owner should reserve the right to himself to purchase all or any of these Items. The Builder should allow for the cost of lifting and fitting-up on board in his "Direct Outlay" Estimate.

Anchors and Cables
Boats
Capstans
Collapsible Gates
Compasses & Nautical Instruments
Davits
Deck Covering Compositions
Decoration of Public Rooms
Disinfecting Plant
Electrical :—
Generating Plant
Lamps and Fittings
Wiring
Bells
Clocks
Telephones
Motors and Fans
Filters
Fire Indicaters
Floor Coverings
Furniture
Galley and Pantry Outfit
Gymnasium Outfit
Hardware
Laundry Outfit
Life Jackets
Lifts
Mattresses
Medical Outfit
Musical Instruments
Refrigerating Installation
Safes
Sanitary Outfit
Steam-heating
Steering Gear
Submarine Signalling
Thermotanks
Telegraphs (Mechanical)
Ventilation, Trunking and Cowls
Water Services (Hot)
Watertight Door Gear
Winches
Windlasses
Windows and Sidelights
Wireless Telegraphy

BUILDERS' PROFIT.

An agreed percentage of the sum of—
(i) Direct Outlay Estimate agreed upon between Owners and Builders. This amount to include a margin equal to (say) 5 per cent. of Direct Outlay Estimate.
(ii) An agreed percentage of the above sum to cover the Establishment Charges.
(iii) The actual total sum allowed by the Owners to cover the cost of the Scheduled Items.

OWNERS' INCIDENTAL EXPENSES.

CERTAIN CONSUMABLE STORES

Excluded from the specification, which by reason of their being of special design, or their being stock articles, the Owner may prefer to supply himself. Should he desire these articles to be supplied by the Builder, he would have a prime cost value assigned to each, and include them under the other scheduled items.

EXAMPLES.

Blankets
Bolsters
Charts
Chronometers
Crystal
Cutlery
Counterpanes
Earthenware
Flags
Linen
Mattresses
Napkins
Plated Ware
Sheets
Silver; Towels, and any similar items.

SALARIES AND TRAVELLING EXPENSES

Incurred by the owner's staff of technical officers, inspectors, and others in connection with the design and construction of the ship.

DISCUSSION.

Professor Sir JOHN H. BILES, LL.D., D.Sc. (Vice-President): My Lords, Ladies and Gentlemen, this paper upon which I have been asked to open the discussion is, in its subject, a disappointment to many of us. I am sure you will quite agree that, from the professional side, we should have been much more interested to have heard Mr. Peskett's views on the question of the design of ships from the shipowners' point of view. Mr. Peskett occupies almost a unique position in having been associated, on behalf of the shipowners, with the designs of some of the largest and most important ships upon the ocean, and, therefore, his views on that subject and the experience which he has gained in carrying out this work, are of the greatest interest to the whole of us. I venture to hope that while we are at the moment disappointed, it is only a pleasure deferred, and that when Mr. Peskett has had the whole winter in which to collect his thoughts and get rid of a large amount of the other work he has in hand, we shall have, at the Spring Meeting, a very full paper on the subject of ship design from the shipowners' point of view.

The subject of the present paper is one which touches very closely the heart and the pocket of the shipbuilder. I have been for some years one of those to whom Mr. Peskett refers as an ambassador between the shipowner and the shipbuilder, and that office is sometimes a very delicate one. I know that in many cases the shipbuilder is not getting as much as he ought, but there is no help for it, because the other shipbuilders are quite willing to incur just as large a loss as he is. That, of course, arises out of competition. Mr. Peskett, very wisely, I venture to think, deprecates the complete elimination of competition by anything in the nature of a trust or a combine, but the steps which he advocates in the paper are really intended to very largely eliminate the disadvantages that are incidental to too severe competition. The difficulty of totally eliminating competition does not seem to have been dealt with here, and, as far as I can understand the intention of the paper, it seems to be that in the first place you should select your builder; having selected your builder, you then arrange on a system somewhat similar to what is described in this paper, to determine what should be the price of the ship and the actual sum to be finally paid for the ship. That is a method that has been developing in recent years, but it is associated in some cases with an attempt at a preliminary competition between shipbuilders, who are asked to take the plans of the shipowner and give a price for a ship based on those plans, and the builder who gives the lowest price is generally looked upon as the one with whom it is best to place the contract. When the price has been fixed in competition in that way, then some more or less successful attempt is made to show that the proper way is to build upon some such system as is shadowed here, on the distinct understanding that under no circumstances shall the price exceed that which was the lowest price of the lowest tenderer. In that way the conditions of competition are introduced and, at the same time, you have the benefit of this contract system indicated by Mr. Peskett, in which the builder and the owner, or their representatives, are not continually pulling in opposite directions. They are not placed in a position where everything that the builder can save from putting into the ship goes into his own pocket, and everything the owner can get for nothing out of the shipbuilder he gets, as he thinks, for the benefit of the ship. Instead of that the two work side by side; they consider jointly what is the best thing to do; they agree that in the end the shipowner has to pay, and they decide what is best in the owner's interest. To that extent this system is one which commends itself to a great many people. I understand from the paper that Mr. Peskett suggests that a price should be agreed between the owner and the builder by calculation. Now that is done in the system which I have just attempted to describe, the difference being that instead of the shipowner and the shipbuilder combining to make an estimate, the shipbuilder makes the estimate in the ordinary way, which is much

more detailed than what is shown here, and the shipowner takes the shipbuilder's estimate as the proper one from which to start.

There is one interesting point with reference to the scheduled items—things that are not included in the first place in the estimated price of the ship proper : the decorations, and arrangements of saloons —which are largely artistic, and are subject to the decision of people who have an artistic temperament, and it is found by experience that it is exceedingly difficult even for a shipbuilder who employs people of that kind to make quite sure how much they will spend in carrying out their artistic ideas. The practice of asking the shipbuilder to take the risk of decorating a ship to the satisfaction of a shipowner is one which a great many shipbuilders have objected to, though some still at the present time prefer to take that risk. On the whole, it does seem to be very much better that the part of the ship which is a matter of taste and fancy, and that part which is absolutely and completely determined by the shipowner, or the cost of it, should be met completely by the shipowner.

I think, Sir, while, as I said at first, we must be considerably disappointed for the present at not having had the paper from Mr. Peskett that we looked forward to, at any rate he has given us a most interesting one, which I am quite sure will lead to considerable discussion on the part of those who, on the one side, like Lord Inverclyde, are interested from the shipowners' point of view, and on the other side from those who are interested from the shipbuilders' point of view.

Mr. John Inglis, LL.D. (Vice-President) : My Lord, Ladies and Gentlemen, I had not intended to make any remarks upon this paper, because it seems to me the scope of it is so wide, and touches so many points, that it is a little difficult to determine whether what one has to say is relevant or not. Perhaps I might make some desultory remarks upon certain paragraphs which have caught my eye as I went along. I would remind Mr. Peskett, first of all, that in this part of the world the shipbuilder and the naval architect are generally one and the same person, and there is no ambassador between the two great powers. That may be an advantage, or it may be a disadvantage, but it has to be put up with on the Clyde, as a rule.

With reference to the remarks of the Chairman of Messrs. John Brown & Co., at the launch of the *Aquitania*, as to the formation of a combination amongst shipbuilders, I think we may dismiss any idea of a combination being formed among such an unwieldy body as the shipbuilders of this country. There is such a thing as a combination amongst shipowners, I understand ; they occasionally meet and arrange rates of freight and other matters, and I think there might be a good deal more intercommunication amongst shipbuilders than at present exists, without any disadvantage to healthy trade.

There is a very important allusion at the foot of the first page of the paper : "The shipbuilding markets of the world are gradually altering their attitude with regard to Great Britain, and countries that at one time relied upon her to build their ships are now successfully competing with her in obtaining orders, even from British owners." I happen to have knowledge of a great many offers that have been made from this country during the last two years to important shipowning companies on the Continent. I am not aware that any one of these offers has been successful ; the whole of the work has been placed abroad with Continental builders, the reason being that the successful tenderers on the other side of the North Sea were as a rule 15 to 20 per cent. below the lowest tenders on this side, and in one instance as much as 50 per cent. below the highest. For one reason, the builder in Germany and Holland is able to buy his steel for about £2 or £2 10s. a ton less than we can, and his labour costs him less ; the men perhaps are not such expert workmen as ours—I am sure they are not more expert—but they are a good deal more diligent, and able to work longer hours for lower wages. When the ship-

building industry becomes less brisk than it is at the present time, I am afraid that some of those Continental shipbuilders will be very severe competitors with us for orders even from British shipowners.

At the bottom of the second page Mr. Peskett draws a comparison between the methods adopted in the construction of a ship and the building of a house. I do not think the system that is adopted in house-building is at all applicable to the building of a ship. I do not think we could entertain the idea of an additional wing to a ship, as you might do with a house. There is an interesting instance here of a contract with Messrs. Swan, Hunter & Wigham Richardson, for the building of the *Franconia*, where a rebate of £20,000 was handed back to the owners. I remember a case when four vessels were built on the Clyde, two by one firm and two by another, and the arrangement was that a maximum price was fixed and any saving which could be effected during the building of these vessels was to be handed back to the owners. I think both shipbuilding firms acted with perfect good faith, but while one firm handed back about £10,000 to the shipowner, the other had nothing to hand back, and did not even make the profit they expected. The builder who returned the large sum was probably supposed to be more liberal or more conscientious than the other, but the fact was that he had not estimated so closely, and for that reason there was a margin to hand back. Establishment charges are not dependent upon the direct outlay; establishment charges, as most of us know, have a way of going on whether there is any work in the yard or not, and they have to be met. Neither do I think the cost of steel work can be determined on the basis of weight. The cost per ton of the steel work on a torpedo-boat destroyer would be very different from that on a tramp steamer or on a Cunard liner. I think Mr. Peskett has some idea that with this method the shipowner ought to have a little more knowledge of the cost of the details of construction than he has. There is a French proverb which says, "Tout savoir c'est tout pardonner," so I am beginning to think if the shipowner knew more about the cost of the details of a ship than he does he would be less surprised at the total cost, and perhaps the shipbuilder would receive more toleration when he presented his bill. There are many ways of making contracts for ships. I remember an occasion when my firm was paid the last instalment for a yacht with a blank cheque; I hasten to inform you, my Lord, that we built two vessels for the same owner after that cheque was cashed!

Mr. W. H. Whiting (Member of Council): My Lord, Ladies and Gentlemen, I share the disappointment which Sir John Biles and Dr. Inglis have expressed as regards the limitations of this paper. There is no doubt that if the original intention of the author had been fulfilled and he had given us a paper on ship design, it would have been of still greater interest to us than this one admittedly is.

The paper travels outside the general scope of our discussions, and some have thought it goes so far outside our ordinary work that it is hardly a proper subject for discussion. That opinion I do not concur in. I think the paper raises, amongst other things, at least one important matter of principle, and, in any case, a commercial method of obtaining a ship which professes to facilitate its production is of interest to this Institution. Of course the great thing is, whether the proposed method of ordering or buying a vessel really tends to give us a better and not a worse ship than by existing methods.

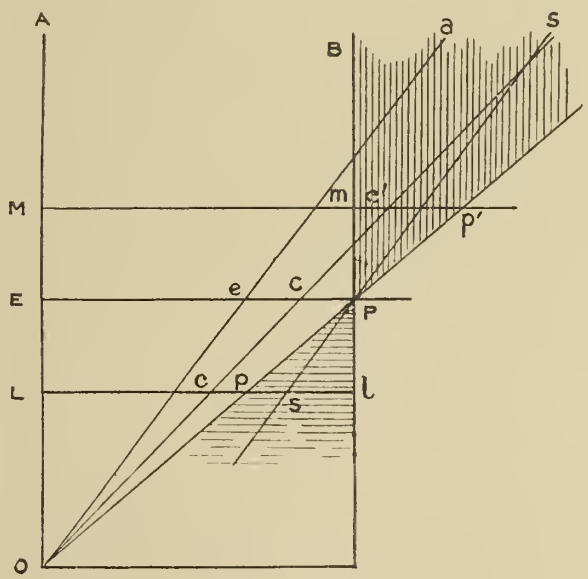
There are two distinct ways of bringing into existence any great engineering production, whether it is a ship, or a bridge, or a house. You may design it before you start, or, on the other hand, you may design it bit by bit as you go along. In the first instance, the vessel is in a sense in existence before a plate is ordered or a rivet put in. Some of the most successful instances of engineering works in this country have been of that kind. The Nasmyth steam hammer stood in Nasmyth's sketch-book, and indeed in his brain, before ever a step was taken to make it in his foundry. The opposite case is illustrated by the man who, after having amassed wealth, travelled far, and seen many architectural

things that he admires, commissions an unhappy architect to embody them in a house, to which he adds bit by bit as he goes along. That, I fear, is a method which, according to this paper, would be applied to shipbuilding. The quantity surveyor, the clerk of works, and the auditor would be brought into our shipbuilding work. It seems to me that it is likely to lead to imperfect engineering results. Of all the types of engineering structures to which to apply methods of that kind, surely the most undesirable is in the construction of those big ships that Mr. Peskett has got in his mind. They are vessels in which, by reason of their speed and size, you are dealing with the most complex engineering problems ; by reason of the routes on which they travel they have to encounter the most stupendous forces of Nature ; by reason of the great cargo of human life which they carry they are running the most appalling risks. To such vessels it seems undesirable to apply methods which tend to facilitate variations of design while building. We are given an example of what occurred in the case of the *Franconia*. The builder handed back to the owners a sum of £20,000 on the sum originally fixed, and this gave great satisfaction to all concerned. I cannot understand how it could be to the satisfaction of all concerned, because either the builder was let off more or less than £20,000 worth of work ; if more, the owner lost, and if less the builder lost. In any case the essential question is, was the £20,000 worth of work wanted at all ? Here is a vessel which is designed to carry £20,000 worth of structure, fittings, and so on, which ultimately she does not carry. Might not a better result have been obtained if these features had never been required ? At any rate, if it is desired to try experimental methods of this kind, let them be tried on a less colossal scale. There are grave questions regarding the type of vessels referred to in this paper which are far from having been thoroughly investigated, and which it would not be possible to open on this platform this morning. It is desirable that we should regard with great caution methods of purchase which, however attractive, would facilitate such variations in design during construction as might well bring about serious results.

The Right Hon. Lord Inverclyde (Associate Member of Council) : My Lord, Ladies and Gentlemen, I had not intended to take part in this discussion : I came here as a listener. It has, however, been pointed out to me that I might speak from a shipowner's point of view, and I do that with considerable diffidence before an audience such as this. I am not sure whether I am the only shipowner present, but at all events we are in a very small minority. Having been associated with Mr. Peskett in a good deal of what he refers to in his paper, I think we ought to understand, and I believe it must have been his meaning, that in advocating this form of contract he referred to that very large class of steamer with which he himself has been associated, and as far as that is concerned I entirely agree with him. It has been our experience that in contracting for these very large liners, steamers which take from one to two years to build, it is really impossible to contract at a fixed price unless the builder makes a very large margin indeed to cover himself for eventualities. In building these large liners we are gaining experience all the time that we are building; and although, as Dr. Inglis truly says, we do not add a wing here and there to a liner, yet when she is contracted for perhaps two years or more before her delivery is due, it is not possible at that date to foresee all that will be required or all the experience we may gain during construction, and which may necessitate changes in the specification. Therefore, as far as that is concerned I am entirely with Mr. Peskett, that this method is a most satisfactory one by which to contract for a steamer, both from the builder's and the owner's point of view. However, I do not think it would be a satisfactory form of contract for tramp steamers, or the smaller class of steamers with which so many of us are acquainted. In regard to the steamers which I own here on the Clyde, and with which I am more intimately connected, I must say that I prefer always to contract at a fixed sum, and in regard to the contracting for these steamers I

confess that I have not taken into consideration, as this paper does, what profit the shipbuilder may desire to make, as I have always felt that I can safely leave that to the shipbuilder himself. We are at the present time in a rather unique position, because of course the yards are very busy indeed, and I confess that there is not amongst shipbuilders that competition to obtain work which I should like to see; but no doubt those things will right themselves in the course of time, and when shipbuilders are not so busy as they are now we will revert to the former state of affairs.

Mr. P. A. Hillhouse, B.Sc. (Member): My Lord, Ladies and Gentlemen, I think it would have been somewhat better if Mr. Peskett had adhered to his original intention and had left in his title the words "from the shipowner's point of view," because I think throughout this paper he has viewed the matter very largely from that standpoint. In the first place, he has denied, as has already been pointed out, the right of combination to the shipbuilders. They are not to be allowed to combine so as to ensure that they shall have reasonable margins of profit, but Mr. Peskett reserves to the owners the privilege of having international meetings to keep up freights and rates of passage for the poor emigrant. Then he has entirely ignored the great risks which the shipbuilder has to undertake. The shipbuilder is always fighting against the laws of Nature, which are inexorable. He has to produce a ship which will carry a given weight, attain a certain speed, be satisfactory from the stability point of view, and be strong enough and sufficiently seaworthy. If he succeeds, he gets (perhaps) a certain fixed margin of profit. If he fails, he has to submit to heavy penalties, and may even have the ship thrown back upon his hands to sell to other people as best he can. Lastly, although Mr. Peskett has dwelt upon that rosy state of things where, upon the completion of the ship, the cost turns out to be less than the estimate, he has entirely ignored that gloomy region in which the actual cost exceeds the contract price. I can illustrate this, perhaps, by means of a simple diagram. The horizontal ordinates of the sloping line O *a* measured from the vertical O A represent the actual cost of the ship, while O E and E *e* each represent the estimated cost. Now it is proposed to add to this a certain percentage to cover charges—these are shown by intercepts such as *e C*, between the lines O *e* and O C—and another percentage to represent profit—shown by intercepts such as C P. The distance E P then represents the maximum sum which the shipowner expects to pay. Now in ordinary contracts the price is a fixed sum, and if the actual cost comes out less than the estimated cost (as shown by the line L *l*), then the shipbuilder gets his definite percentage profit (*c p*) and, in addition, a sum represented by the distance *p l*; but the area here of possible advantage to the shipbuilder, as shown by horizontal shading, is a restricted area. If, on the other hand, the cost exceeds the estimate, as shown by the line M *m*, then the shipbuilder lays out an amount of money represented by M *c'*, but receives only the amount represented by M *m*, thus losing not only the profit *p'c'*, but also a part *c' m* of his actual outlay. The whole of the vertically shaded area represents loss to the shipbuilder, and this area is unlimited in extent. Now a fair thing would possibly be to agree that,



whatever the cost, the builder would be paid a sum represented by the outside sloping line O P, that is, that he would always get a definite percentage for charges and for profit. The whole of the horizontally shaded portion, which formerly went into the builder's pocket, would now be retained by the owner, while the vertically shaded part, which formerly represented loss to the builder, would now become extra payment by the owner for extra value received. That would be a fair kind of thing to do, but it would have this disadvantage, that it would offer no inducement to the builder to reduce the cost. He would always be sure of his definite percentage, and it would give him an inducement to increase the cost, because he would thereby increase the actual amount of his profit, although not the percentage amount. Mr. Peskett proposes another line, which is in effect the line P S, drawn parallel to the cost line O *e*, so that in the event of the cost being less than the estimate the builder would get his definite percentage of profit *p c*, and in addition the amount *p* S, while the owner would get the amount S *l* ; that is to say, the builder, while getting no more actual profit than he expected, would obtain a larger percentage on his actual outlay, while the owner would pay less than he expected. I have no fault to find with such an arrangement. It is a very reasonable proposal, but, unfortunately, Mr. Peskett's line deviates from the path of justice at the point P, and instead of carrying on in the same direction to S′, it suddenly turns and goes upwards to B, so that, although he allows the owner the advantage of any possible saving below E P, he does not give to the builder any compensation if the cost of the ship exceeds the contract price. I would suggest that it would be fairer to continue Mr. Peskett's line without a break into the region of possible loss, so that in the event of the cost exceeding the estimate, the loss would be shared in some proportion between the builder and the owner, just as, in the opposite event, the saving would be shared in some definite proportion between them.

Sir Archibald Denny, Bart., LL.D. (Vice-President) : My Lord, Ladies and Gentlemen, I should like to ask a question before Mr. Peskett replies. Supposing the rate of wages rises during a contract, such as Lord Inverclyde has referred to, which might last for two years, is the estimated cost also increased by that amount ? That is a point, I think, which is not quite clear in the paper. I gather from Mr. Peskett that the estimate made is an agreed estimate between the builder and the owner ; the builder and the owner both check the estimate and come to a conclusion it is a reasonable one, but if there is the proviso that if wages increased unexpectedly there should be an addition, that would, I think, very considerably modify the criticism which has been directed against this paper.

Mr. L. Peskett (Member) : My Lords and Gentlemen, I do not think I have anything to say beyond thanking those who have taken part in the discussion. The justification of proposing a standard method of preparing estimates may be found by reading shipbuilders' balance sheets, and the tenders which have been received during the past few years by the Cunard Company. For a steamer 600 ft. by 72 ft. by 47 ft. (moulded dimensions) the difference between the highest and lowest tender was £100,000, and for ships of much less dimensions the difference has amounted to nearly £50,000.

The Cunard Company, and most of the leading shipowning companies, prepare their own plans and specifications before sending out for tenders, and there is no difficulty in preparing an estimate on the basis outlined in my paper. Should any additions to the vessel be found necessary during the progress of construction, an estimate of the extra on contract price is mutually arranged, the extra including a *pro rata* sum on profit and establishment charges. This, I think, covers the remarks made by Mr. Hillhouse and Mr. Whiting.

With regard to Sir Archibald Denny's question, I think that is dealt with in the paragraph next to the last in the paper : " The cost of meeting the requirements of post-contract legislation is obviously

the subject of an extra-contract agreement between owner and builder." I do not know whether that answers the question ?

Sir ARCHIBALD DENNY : No, not unless you count Trade Union action as post-contract legislation.

Mr. PESKETT : I think it would come under that heading.

The PRESIDENT : Ladies and Gentlemen, in rising to ask you to give a vote of thanks to Mr. Peskett for his paper, I am very glad that the discussion we have had has drawn from him the statement that he will give us a paper on ship design at our next meeting. I ask you to give a hearty vote of thanks to Mr. Peskett for his paper.

ON SAFETY OF LIFE AT SEA.

By PERCY A. HILLHOUSE, Esq., B.Sc., Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 24, 1913; Rear-Admiral the Most Hon. the Marquis of BRISTOL, M.V.O., President, in the Chair.]

HUMAN life at sea is on the average exceedingly safe. The stories of great shipping disasters are of so great a dramatic interest and attract so much attention that the comparatively large number of persons carried safely is apt to be overlooked. Taking the records of the North Atlantic as example, recently published figures show that during the twenty years 1892-1911, some 95,000 voyages were made between Britain and America, about 350,000 crew and over 9,390,000 passengers having started. Out of this large number of trips, accidents involving loss of life occurred in only 165 cases, 1,057 crew and 80 passengers having been lost. Only one out of 332 of the crews and one out of 117,400 passengers did not reach their destination in safety. Of the 1,137 souls which so perished, 39 were lost in 7 cases of foundering, 187 in 10 strandings, 9 in 6 cases of collision, 195 in 113 cases of accidents on board due to heavy seas, fire, explosion, bursting of steam pipes, &c., while no fewer than 707 were lost in 29 vessels which were posted as "missing." Owing to the large proportion of losses in missing vessels, it is impossible to obtain a definite analysis of the causes of death, but if we assume that of the 707 missing, 400 were in 16 vessels which foundered, and 307 in 13 vessels which sank after collision, either with other vessels, derelicts, or ice, we obtain the following percentages :—

Cause.	Number of accidents in percentage of total number of accidents.	Lives lost in percentage of total loss of life.
Foundering	14	39
Stranding	6	16
Collision	12	28
Accidents on board	68	17
	100	100

About two-thirds of the total number of casualties are thus seen to be of a minor nature, involving comparatively small loss of life, while the remaining one-third accounts for 83 per cent. of the total loss.

Life may be endangered at sea either by accidents occurring on board, from which the majority of the vessel's complement escape, but to which a few fall victims, or by disasters resulting in the loss of the vessel, together with all or a portion of her passengers and crew. Among the former minor class we may enumerate fire, explosions of gas, bursting of steam pipes, breaking loose of cargo, accidents to machinery, emanations of poisonous gas from certain cargoes, and heavy seas breaking over the vessel, sweeping her decks and damaging her deck-houses or fittings. The safety of the whole ship's company is jeopardised by stranding, foundering, capsizing, or by collision with other vessels, derelicts, floating ice, or mines.

It is from the second category that the greatest disasters arise, and though such calamities are fortunately few in number, yet they are usually accompanied by a comparatively large loss of life. It is, therefore, the duty of the naval architect and shipowner to devise and adopt all the means in their power, firstly, to avoid such accidents, and secondly, to afford means of saving life should such accidents unhappily occur.

Much has already been done to minimise these dangers. The general introduction of steam power, the preparation of accurate charts, the better lighting of coasts, improvements in the mariner's compass, the study of the earth's magnetism, fog horns, submarine sound signalling, and wireless telegraphy have robbed the coast line of many of its terrors.

International regulations as to the Rule of the Road at sea and as to the lighting of ships, the systematic reporting and destruction of derelicts, reports as to the presence of ice and the introduction of wireless telegraphy and sound signalling have made collisions at sea comparatively rare.

Probably very few or no vessels actually capsize at sea, as the problem of stability is now so well understood that any signs of tenderness can be taken in good time and corrected by the use of water ballast. It is, however, conceivable that sudden capsizing at sea might occur if, through stress of weather, large quantities of water found their way into the holds, and so greatly reduced the effective moment of inertia of water plane. Such cases, however, should perhaps be classified rather under the head of foundering than under capsizing. Of all the causes of loss of shipping, that over which the naval architect can probably exercise the greatest control is that of foundering, whether through stress of weather or as the result of collision with any obstacle whatever. The proper regulation of freeboard has a very important bearing upon such disasters. Statutory freeboard is designed to ensure a suitable proportion of reserve buoyancy, and takes into consideration the type of vessel, her strength, and the extent of her erections. In most cases it determines the height of the watertight bulkheads

above the load waterline and this margin, together with the spacing of the bulkheads longitudinally and the water-excluding properties of the machinery, the coal and the cargo carried, are the elements which determine the amount of flooding which can be endured without submerging the tops of the bulkheads and so leading to total sinkage. In most large passenger vessels of modern type the number and extent of the erections above the bulkhead deck affords also a height of platform above water which greatly reduces the danger of structural damage and loss of life due to seas finding their way on board.

Subdivision of the hull into separate watertight compartments is, in conjunction with the provision of a suitable freeboard, the most valuable means at the disposal of the naval architect towards limiting the amount of sinkage consequent upon any given extent of damage to the shell plating. In war vessels subdivision can be, and has always been, carried out to a much greater extent than is possible in the case of merchant craft. Minute subdivision in merchant vessels would interfere too much with the cargo and passenger spaces, and so render them commercially impossible, but without going to this extreme much may be done towards the ideal of an "unsinkable" ship. An absolutely unsinkable ship is impossible, unless the average density of hull, machinery, and loads is less than that of water. In the days of wooden sailing vessels this may have been realisable, but not in the case of modern vessels built largely of steel and carrying great weights of machinery and boilers. All that can be done is to limit the space into which water can find access, so as to reduce the amount of sinkage due to any assumed damage, and to ensure that the bulkhead deck will still be well above water. Limitations to the extent of flooding may be imposed by transverse, longitudinal, or horizontal divisions, or any combination of these. It is interesting in this connection to find in Volume II. of our Transactions a paper read by Mr. Charles Lungley in 1861, strongly advocating the fitting of watertight decks in order to increase the safety of vessels in case of bilging accidents. Until comparatively recent years subdivision was obtained mainly by means of transverse bulkheads, and the number of these was determined by the regulations of the Board of Trade and the Classification Societies. In most cases the resulting subdivision was not enough to enable the vessel to remain afloat with more than one compartment flooded, and it was found difficult to comply with the "two compartment" standard of the Bulkhead Committee of 1891. In many cases it was found that the interference with cargo and passenger arrangements was too real, and the increased safety in case of accident too remote, to make the proposal sufficiently attractive, even although some reduction in life-saving appliances was permitted; and many ships begun on a "two compartment" basis were finally completed upon ordinary lines. But the problem is again being seriously considered; a second Bulkhead Committee is sitting, and its Report will be of vital interest to the whole shipbuilding and shipowning community.

Increased safety may be obtained by means of transverse bulkheads in two ways. Firstly, by increasing their height so that the vessel may sink more deeply before their tops become submerged. This was done in the celebrated *City of Paris* and *City of New York*, and is equivalent to increasing the freeboard to the bulkhead deck. The method is apt to interfere with the passenger accommodation, and unless many watertight doors are fitted results in much inconvenience in the working of the ship. It is largely a question of convenience *versus* safety. Secondly, by decreasing their spacing, fitting more bulkheads, and so reducing the sizes of the holds and 'tween decks. This also tends to interfere with cargo and passengers, and increases the numbers of cargo hatches and stairways required. It should be noted in this connection that it is not so much a question of the *number* of adjacent compartments that may be simultaneously flooded with safety as the *proportion* of the ship's length which may suffer damage. If a ship's side be ripped open for any given length it is of little consequence into how many compartments that length may be subdivided. All will be laid open, and the result will be the same whether the vessel be a "two compartment" or a "three compartment" ship. When a ripping blow occurs the *average* length flooded will be equal to the length of the rip plus the length of one compartment, and the *maximum* length laid open will be when each end of the rip lies on a bulkhead. In this case the length flooded will be two compartments more than the length of the side cut open. The smaller the compartments, therefore, the less will be the amount of flooding, and to this extent only is a "three compartment" ship likely to be better than a "two compartment" design if the extra safety is obtained only by closer spacing, and not also by increased height of bulkhead.

In cases of collision with another vessel, in which it may fairly be assumed that only one transverse bulkhead will suffer damage, and consequently not more than two adjacent holds be flooded, the vessel having the closer bulkhead spacing will, of course, have the advantage, since a less volume of water will be admitted.

Great care requires to be observed when introducing longitudinal bulkheads as a means of limiting the extent of flooding, on account of the fact that water admitted to one side of the ship only will produce a heeling tendency, which may have the effect of bringing the tops of the bulkheads upon that side nearer to the water surface than would be the case were both sides simultaneously open to the sea. In spite of the loss of water plane area, the vessel's stability in the flooded condition will usually be greater than when intact, on account of the increased height of the centre of buoyancy, and this fact is of much value in reducing the amount of heel produced by unsymmetrical flooding. Longitudinal subdivision is commonly obtained either by means of middle line bulkheads in machinery and cargo spaces, by wing bulkheads in engine rooms, and between boiler rooms and side bunkers, or by carrying the double bottom well up the

vessel's sides so as to form an inner skin. Of these three methods, that of wing bulkheads appears to be the most valuable, as the wing spaces are less than those on one side of a central bulkhead, and greater than those between two skins. The central spaces containing the machinery and boilers are less likely to be rendered useless and the wing bulkhead, being further from the vessel's side than the inner skin, is less liable to partake of any damage occurring to the outer shell plating. In cases where wing bulkheads are fitted it will probably be found advisable to interconnect the wing compartments on either side of the ship, so that both may be filled at the same time, and the vessel thus kept upright.

There are many objections to the fitting of an inner skin in mercantile vessels. Wing bulkheads already serve a useful purpose as bunker boundaries, and so do not add materially to the vessel's weight or cost, but an inner skin is a definite addition to both, it reduces the space available for cargo, coal, machinery, and passenger accommodation, and on account of its comparative nearness to the ship's side is difficult to dissociate from any damage occurring to the outer skin.

Watertight decks form an exceedingly valuable form of subdivision. Very little additional weight is involved, as the deck is already required for other purposes. If the deck is above the waterline and between passenger accommodation and cargo spaces its conversion into an effective watertight division is comparatively simple, and the only alteration involved is that of enclosing the cargo hatches by watertight trunks extending up to the bulkhead deck. If, however, there happens to be passenger accommodation below the watertight deck it becomes necessary also to carry watertight exit stairways up to the bulkhead deck, and in most cases this would mean a considerable amount of inconvenience to passengers. Watertight doors may of course be fitted on such stairways, but these should be avoided as far as possible since every additional opening affords passage to water should the closing of the door be overlooked or found impossible in case of sudden emergency.

In considering the position of the waterline to which the vessel is estimated to sink after any supposed flooding, special attention requires to be given to the possibility of water entering non-damaged compartments by way of scupper pipes, slop shoots, sanitary discharge pipes, and so on. The storm valves fitted where such pipes pass through the vessel's sides are not absolutely watertight, and should the upper ends of the pipes fall below the external sea level, water may easily find its way inboard. Screw-down covers should be fitted in all such cases, and as the rate of flow would not be great, time would be available for the closing of each pipe before much additional water had entered the vessel.

The whole success of the method of subdivision hangs also upon the prompt and

efficient closing of all watertight doors as soon as an accident has occurred. There are now upon the market several excellent systems by which all doors can be closed from the bridge, and in which any door is closed automatically by the rise of water on either side of the bulkhead.

In the last resort, when all else has failed, and it becomes evident that it will be necessary to "abandon ship," the safety of life at sea will depend upon the amount and nature of the "life-saving appliances" with which the vessel is equipped.

It is now a generally accepted axiom that lifeboat accommodation should be provided for every person on board. In April, 1912, all cargo steamers actually carried "boats for all" on *each* side of the ship, and 80 per cent. of foreign-going passenger ships were also provided with boat accommodation for all on board.

Corresponding figures are not available in the case of vessels employed in the home trade, but it is worthy of note that in 30 out of the 35 casualties which have occurred to home trade passenger vessels during the 20 years 1892-1911 there was actually boat or buoyant deck-seat accommodation for all on board; and that in the remaining reported cases it did not appear that further life-saving appliances would have been of any use. Three-quarters of the total loss of life occurred in three disasters to vessels in which the boat accommodation was in excess of the number of persons on board at the time of the accident.

"Boats for all" is excellent in theory but exceedingly difficult to realise in practice. The *Olympic* now carries no fewer than 68 lifeboats, and the *Aquitania* will have 92. The problem of finding deck space for such large numbers of boats, each measuring about 30 ft. in length and 9 ft. in breadth, and weighing about two tons, is one of great difficulty, exceeded only by that of devising satisfactory means of filling them with their proper complement of passengers, and thereafter lowering them safely into the water. The problem becomes still more complicated if it is further attempted to arrange the boats so that all can be lowered away from either side of the parent vessel. Much ingenuity has been expended upon the problem, and many forms of boats and davits have been devised. Most of these are of too complicated a nature to be of practical value. It is just possible that the provision of large numbers of boats, many of which have to be stowed one above another and not attached to davits, may be worse than useless, as the extra boats may so hamper the deck space that access to the boats under davits will be hindered, and it may not be found possible to free the inboard "nested" boats from their supports and lashings, to attach them to davits when their turn comes, to load them and lower them away, in sufficiently short a time. It is probable that in most cases inboard boats would be more likely to serve their purpose if arranged so as to float off the sinking vessel and act as life-rafts to be reached by swimming.

The ideal life-saving appliance is one that is simple and certain in its action and by which large numbers of people can be floated safely and quickly. Floating decks and floating deck-houses have been proposed, but no solution on these lines has so far appeared to be of sufficient merit to meet with general approval. The latest trend of expert opinion appears to be towards the adoption of a small number of large lifeboats, rather than the provision of a large number of smaller and less substantial craft. The whole subject has been under the careful consideration of the Boats and Davits Committee, and their final report has recently been made.

Briefly put, its most important recommendations are as follows :—

The number of persons to be allotted to any open lifeboat is to depend upon its stability and upon its actual capacity as determined by Stirling's Rule, instead of upon the overall dimensions and an assumed coefficient.

Development may safely be in the direction of larger boats, say 50 ft. by 15 ft. by 6 ft. 8 in., each carrying 250 persons and weighing 28 tons when loaded.

Decked lifeboats to have permanent bulwarks fitted with buoyancy tanks: they may be "nested" two or three deep, even although it may be necessary to raise them out of the nest by means of hand winches before they can be moved under davits.

Pontoon rafts may be accepted for 25 per cent. of the persons for which the ship is certified.

In foreign-going ships the lifeboats are to be transferable to either side.

Davits to have gearing to turn the boat out even against a considerable list. To have non-toppling blocks or two sets of falls if they deal with more than one boat, and wire falls with lowering drum and brake if with more than two boats.

Motor boats to be optional: in any case not more than two will be necessary on each side of the vessel: their radius of action to be 100 miles on paraffin fuel.

It remains to be seen how soon, and to what extent, these recommendations will be embodied in Statutory Rules, but there can be no doubt that they would have an important effect upon the problem of boat accommodation and stowage. The adoption of Stirling's Rule and some definite (but yet to be determined) standard of stability will tend to produce a form of lifeboat capable of carrying more persons with safety on any

given length, breadth, and depth than the forms at present in vogue. This will reduce the number of boats required, and so facilitate stowage. The provision most difficult to carry out in practice will, I think, be that of arranging that the boats shall be transferable to either side of the vessel. The central portions of the boat deck of the modern first-class liner are already so much occupied by saloon domes, officers' houses, ventilators, water tanks, funnels, masts, and stays, that it will be a matter of some difficulty to find one or two clear spaces across which a lifeboat 30 ft. to 50 ft. in length can be moved. As each boat may have to be moved longitudinally, in order to come abreast of the cross gangway before transference to the opposite side of the ship, a good deal of "shunting" will have to be done to clear the davits and other deck obstructions. Care will also have to be taken that there may be no possibility of a heavy lifeboat and the truck upon which it is being moved "taking charge" if the vessel is rolling in a seaway.

The Committee has done much painstaking work in testing the stability of open lifeboats, decked lifeboats, deck-seats, and rafts. Much light has been thrown upon a subject hitherto greatly neglected, and the results of their labours will undoubtedly be to encourage the development of more reliable and stable forms of all such appliances.

But when all has been said and done I think it is certain that in the future, as in the past, safety of life at sea will depend more upon the avoidance of accident than upon its repair, and that careful navigation will prove of infinitely more value than the most minute subdivision or the most perfect equipment of life-saving appliances.

And though an increased immunity from loss of life at sea may and will be recorded in the years to come, yet it must never be forgotten that life cannot be made *absolutely* safe at sea any more than upon land. "They that go down to the sea in ships, that do business in great waters" do so at risk of their lives: we may minimise, but cannot annihilate that risk.

DISCUSSION.

Sir Archibald Denny, Bart., LL.D. (Vice-President): My Lord President, Ladies and Gentlemen, I think the members will sympathise with the Chairmen of the three Committees who are present to-day, Sir Philip Watts, Sir John Biles, and myself, and although Sir John Biles' Committee has reported, he also has a little hesitation about speaking on this paper, because there is the international position to consider. Perhaps you do not know there is to be an International Conference upon this question of safety, and it would not become the Chairmen of the Committees to discuss this question

at any great length. Therefore I propose simply to suggest a very hearty vote of thanks to Mr. Hillhouse for his paper. I think he has covered all the ground which the three Committees are considering. Perhaps I might venture so far as to make this remark: he has not expressed a very definite opinion as to what he thinks about the large lifeboat; but I must say, the more I consider the problem of the large unsinkable lifeboat, decked all over, into which you can put the people and shut them down, and either launch her into the sea or allow her to float off if you are too late to launch her, the more it appeals to me. Then Mr. Hillhouse says that in large vessels of this kind, in which these boats would be fitted, it is difficult to find a space across the deck which is not already encumbered by deckhouses, domes, &c. That is just the point, Gentlemen. If the regulations are laid down you have to find the space and suppress the domes and deckhouses, and there is an end of it. You cannot have your cake and eat it too.

A Belgian authority, I think, put the question of safety at sea very well. He said the three great life-saving principles were, wireless telegraphy, subdivision, and boats—wireless to call help, subdivision to keep the vessel afloat until help arrived, and boats to transfer the people to safety. Now, were it not for the risk of fire, you might perhaps have said that subdivision would meet the question of safety so far as the passengers were concerned; but you always have the danger of fire, and I do not think there is a person in this room who would face the possibility of leaving a lot of people on board a ship on fire to be roasted to death: I think you would sooner see them drown. "Boats for all" is, therefore, not a parrot's cry, but is based upon very sound principle.

I feel it is absolutely impossible for me to say much: I would like, however, to remark that upon page 20 Mr. Hillhouse refers to watertight decks, and says that very little additional weight is involved. I do not know that that is strictly true under all circumstances. You can conceive of a watertight deck fairly low in the ship which, were a long length of the vessel ripped open and the vessel changed her trim and her immersion very considerably, might be under a very severe pressure: you might have 20 or 25 ft., or even more, of upward pressure on that deck, and I think it would be worth while making the calculation. If Mr. Hillhouse would do so, I am quite sure my Committee would be delighted to have evidence from him on that very interesting point. We are still open to take evidence.

I am also obliged to Mr. Hillhouse for drawing attention to Mr. Charles Lungley's paper. Mr. Hillhouse's remark about the unsinkable ship is perfectly true, and he knows all the difficulties of the subject. If he had been on the Committee, he would have found, as I did, that non-technical men found no difficulty whatever in making a ship unsinkable. You had only to fill her with airtight tin boxes, and there she was; even if you only filled the deck between the beams, whenever the ship came down to that deck and the tin boxes came on the water, she could not sink any further, she would be floating on air! These people did not have that clear conception of what displacement and weight are which Mr. Hillhouse has. After all, whatever the Committee may report, and whatever the Board of Trade may decide, and whatever the nations may decide internationally, there is no doubt that careful navigation is the prime factor for ensuring the safety of passengers at sea.

Mr. S. J. P. Thearle, D.Sc. (Vice-President): My Lord, Ladies and Gentlemen, I find myself in precisely the same position as Sir Archibald Denny. As a member of the Bulkheads Committee, I do not feel free to discuss this paper, and, therefore, I do not propose to do so. I can only express my high admiration of the quality of the paper, which I have read with great attention and respect. I shall continue to refer to it until our work as a Committee is completed.

Mr. H. M. Napier (Member): My Lords, Ladies and Gentlemen, I must in the first instance compliment Mr. Hillhouse upon the beautiful composition of his paper. We look on Mr. Hillhouse

as a scientific man on the Clyde, but he is a man of many parts. This subject of life-saving at sea is one which has been in everyone's thoughts since that great disaster in the Atlantic, and it has brought forth advice from all sources on what is to be done in future. From this paper it appears that the *Aquitania* is to carry 92 lifeboats. To my mind that is a ghastly state of things. What could possibly happen in the event of these 92 lifeboats being suddenly required for use ? It sounds very much like the story of the Bishop going to the Captain during a great storm and being told he must trust to Providence !

With regard to the question about shifting boats from place to place, or from one side to the other, as Sir Archibald Denny has said that it is most essential that deckhouses, skylights, and other things should be simply cleared away and that skids should be fitted. No big vessel goes down in the Atlantic, or very very few, without having a bad list to one side or another, and it is usually absolutely impossible to get large boats to the lee side, *i.e.*, to the low side, on account of deck erections, and it is equally impossible to launch them on the high side. There must be skids to shift them from side to side. I remember a good many years ago building for a man who was very keen to fit deck skids right across, but he would not clear away his engine skylights to do it, neither would he pay the expense of putting heavy skids on the top of them, so the skids were not fitted.

We are greatly indebted to the Committee which has dealt with the question of life-saving appliances, and we all look to it for something in the future, although it must not be forgotten that life is a compromise from beginning to end ; we must take some chances on board ship. I am not touching upon the subject of watertight subdivisions. That is the first thing, as Mr. Hillhouse has said, much more important than boat accommodation, and the Transactions of the Institution of Engineers and Shipbuilders show records of cases where good bulkhead arrangements have saved a ship from foundering after being pierced.

Professor W. Hovgaard (Member) : My Lords, Ladies and Gentlemen, this excellent paper of Mr. Hillhouse covers such a large ground that I can only offer a few remarks on one of the subjects treated in the paper, namely, that of watertight subdivision of ships. In discussing that subject one has to bear in mind clearly what are the principal risks that a ship incurs. They belong to two classes : one is foundering, running ashore on the rocks, which in particular is likely to damage the lower part of the ship, under the bottom ; the other is collision, whether with other ships or with ice or other obstructions which are likely to damage the ship at or near the waterline. It is very important to bear in mind that distinction between the two forms of damage, because they are separated by a belt which is less liable to damage about the mid depth of the ship. Another point of importance to remember is that it is the fore body of the ship in particular which is liable to all kinds of damage. If we bear these points in mind it is quite clear that we ought to provide, speaking of the largest class of ships, and particularly the large liners, first of all with a double bottom in order to provide for damage to the bottom. That is already done in well-designed ships. Then, coming to the sides, we ought to provide for damage by doubling the sides of the ship in some form or other. The author says that the inner bottom is not so well suited to this purpose as a wing bulkhead, and I quite agree with him, for the inner bottom is likely to share the damage to the side. A wing bulkhead may be of two kinds ; it may be an intact wing bulkhead, placed 6 ft. or 7 ft. inside the outer shell, absolutely intact, making a cellular space between the bulkhead and the side ; or it may be a bulkhead which serves at the same time as a bunker bulkhead, but then it must necessarily have doors in it. I think the wing bulkhead, the first form, is decidedly to be preferred, because it can always be relied upon to be intact.

Now, coming back to the fact that the mid depth of the ship is less liable to damage than either the

bottom or the waterline region, that leads to the conclusion that there ought to be fitted a watertight deck at about the half depth of the ship. That will in most cases be possible, without interfering too much with the service of the ship. You will find, for instance, in ships like the *Olympic*, a deck covering the cargo spaces forward, which is about in the position I have in mind. That deck could, I believe, without great inconvenience to the service of the ship, be made watertight. It has the advantage that, if the ship suffers bottom damage, that deck will form the upper boundary for the flooded portion, and only a limited amount of water will be able to enter the ship. If the ship suffers damage in the waterline region, whether by ice or by collision, then the same deck will form the lower boundary of the flooded region. In both cases the amount of water entering the ship will be considerably limited. Such a deck will be of particular importance in the fore part of the ship, because, as I said before, this is the part most exposed to damage. It may not be necessary, perhaps, to fit such a deck in the after part of the ship.

Another point I wanted to touch upon was the question of the bulkheads themselves. Of course, the bulkheads should be strong enough to withstand the pressure to which they are liable to be subjected. That goes without saying, but still it is a fact that such has not till recently been the case, even in the largest ships. Moreover, they should be, as far as the main bulkheads are concerned, entirely intact; that is, they should have no watertight doors. A watertight door always constitutes a danger under any system of closing. It may not be out of place to tell a story about automatic watertight doors when they were first introduced on Atlantic liners. They were so arranged that when a button was pressed in the chartroom all the doors were closed and an electric lamp would show for each door whether it was closed or not; it would glow when the door was closed. The captain aboard that ship was very anxious to impress upon his lady passengers the great safety which these doors constituted, and he used to invite the ladies up to the chartroom, press the button, and show them how the lamps would glow. One day there was a young lady who was more sceptical than the others, and she said, "Yes, I can see the lamps glow all right, but are the doors closed?" "Yes, of course," said the captain, "or the lamps would not show the light." But her logical mind was not satisfied with that, so finally the captain said "We will go down the ship and I will show you that the doors are closed." Then the whole party went down into the ship, but the first door they met was wide open. They went to the next door. That was open, too, and right through the ship all the doors were open. Then it turned out that the captain had so often practised the closing of the doors in that way that the firemen and engineers had got tired of it, and had arranged the switches so that when the button was pressed the lamps would be lighted without the doors being closed. That merely illustrates that any invention or automatic appliance can fail. It may be interfered with wilfully, by malice or for some other reason, or it may get out of order; but that is not all. Watertight doors may be interfered with in many other ways, as in the *Victoria*, where the bulkheads had warped and certain doors could not be closed. Others were not closed because there was no time to close them. Then there may be obstructions which prevent the closing of a door. In fact, a door must always be a danger, and what I want to point out is, that the main bulkheads in a ship should not have any watertight doors at all, and we ought to accept the inconvenience and the cost which it will undoubtedly entail, to have the bulkheads kept intact.

Mr. P. A. Hillhouse, B.Sc. (Member): My Lord, Ladies and Gentlemen, in reply to Sir Archibald Denny, I may say I think his remarks about the large lifeboat are exceedingly good. There can be no doubt that the problem of fitting these boats and of lowering them away will present many difficulties, but we must face those difficulties and overcome them. I would be quite willing to cross the Atlantic or any other ocean in a ship absolutely devoid of boats, and trust entirely to careful navigation and

NOTE ON SOME CASES OF "FATIGUE" IN THE STEEL MATERIAL OF STEAMERS.

By S. J. P. Thearle, Esq., D.Sc., Vice-President.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 25, 1913; the Right Hon. Lord Inverclyde, Associate Member of Council, in the Chair.]

Evidences of what is known as the "fatigue of metals" have from time to time been disclosed when surveying steel vessels under repair, but during the last year or two certain phenomena of the kind have been observed which possess special features, and it is thought that these may be of interest to the Members of this Institution.

In the first of these cases which came under notice a crack was observed in the shell plating on each side of a vessel a short distance forward of the collision bulkhead. These cracks extended right through the plating, and from the amount of wasting by corrosion which had taken place within the cracks it was evident that they were not of recent date. The cracks were of irregular form and did not break into rivet holes, nor were they near the edges of the plating.

Soon afterwards another such case occurred, except that in this instance the cracks were at some short distance abaft the collision bulkhead. Then another and yet another case came under attention, so that surveyors at certain repairing ports began to look out for them when vessels came under survey.

It was evident from the first to experienced surveyors that the symptoms were similar to those due to fatigue in the steel, such as had been observed at some other parts of vessels.

It was remembered that some twenty years ago similar cracks were sometimes seen in margin plates of double bottoms at the backs of the angle bar attachments of the frame heel brackets. Leaks in the margin plate had led to the removal of the attachment angles, and then it was found that cracks had developed in the margin plate at the backs of these bars, in the vicinity of the uppermost rivet and sometimes near the rivet next to it. Such cracks were rarely joined to the rivet hole, but generally they partially encircled it, and it may be mentioned that they were mostly confined to No. 1 cargo hold.

subdivision, were it not for the danger of fire, an ever-present danger which is not overcome by subdivision. Unless it can be shown that fire can be brought entirely under control, I am afraid we must continue to have boats for all. The problem of deck strength which has been suggested by Sir Archibald Denny is of great importance; so far I have not examined it closely to discover what extra weight may be involved.

As regards Professor Hovgaard's remarks about the intact wing bulkhead, that, no doubt, is ideally a very good thing to have, but as the space between the intact bulkhead and the ship's side is lost space, and can be put to no use, the temptation will be to make it as small as possible by bringing the intact bulkhead close to the vessel's side, so that very soon it would degenerate into an inner skin, and would be liable to partake of damage to the outer shell. What he has said about watertight doors is exceedingly true. It is again a question of safety *versus* convenience. In warships it is the practice to have the main bulkheads absolutely without doors. That leads to inconvenience, but the inconvenience is borne in view of the extra safety, and it would perhaps be well to accept that inconvenience in the case of mercantile vessels, also.

The PRESIDENT: I am sure it will be the wish of everybody present to thank Mr. Hillhouse for the very interesting and able paper which he has given us to-day.

Careful investigation showed that the cracks were the result of fatigue in the material of the margin plates, due to frequent alternating pull and push stresses brought upon the margin plate by the heels of the frames, and continued over a long period of time.

The remedy was found in giving the frame heels a better connection to the double bottom, and this was effected by using double instead of single angle attachments at the margin plates, and by the more extensive use of gusset plates and angles in connecting the frames to the inner bottom plating.

Instances of fatigue at this part of a vessel are not often seen at the present day, the remedies adopted having proved to be efficient for the purpose.

The cases in which the phenomena now under consideration had been observed were, therefore, analysed, and it then appeared that they had certain features in common which pointed to a common cause, and consequently suggested a common remedy.

It will be of interest to mention here that up to the date of reading this note about thirty-three vessels have been found when under survey to have been affected in this way.

These are all steamers, ranging from about 3,000 tons to about 5,000 tons gross register. Their ages, when they came under survey, ranged from two to sixteen years, the average age being about nine years.

They are of all types ranging from single-deck to three-deck, spar-deck, and shelter-deck vessels, and some have web framing, while others are framed with the deep frames, ordinarily adopted at the present day.

There is nothing therefore as regards type of design or system of framing common to the vessels, but all of them are of steel. Moreover, they are the products of seventeen different builders and seven different building ports.

Figs. 1 to 7 inclusive on Plate I. show the bow portions of the profiles of seven of the vessels which may be taken as samples of the total number, and the same figures show sectional views of the vessels in way of the cracks.

On Fig. 1 is also shown a part plan of the vessel which is shown in part profile and in section on the same figure.

Figs. 8 to 12 inclusive on Plate II. show some of the cracks on an enlarged scale, and on Fig. 12 is also shown a sectional view of a crack which has undergone some amount of wasting.

The positions of the cracks found in the shell plating of these vessels are indicated on these sketches.

Upon referring to these sketches certain features will be apparent, and these it may be remarked are common to all the cases.

These features are as follows:—

(1) The cracks are always found either in the fore peak or in the fore part of No. 1 cargo hold.

(2) They always occur at the back or at the edge of a chock angle attachment of a side or panting stringer plate to the shell plating.

(3) They always occur in close vicinity to a frame unsupported by a panting or other beam, and they never occur in the vicinity of a frame which is supported by beams.

This statement suggests at once the cause and the remedy.

The primary cause is evidently slight local movement, frequently repeated and extending over a long period of time; and this slight local movement is also evidently due to something which distinguishes the frame at which it occurs from the adjacent frames; that is to say, in being unsupported.

It will be observed by referring to the sketches (which show actual cases in every detail) that there cannot have been any panting in the structure itself, for the number of tiers of beams and panting stringers quite preclude such from taking place. The slight movement referred to has occurred only at the intermediate frames between those to which the beams are attached.

Note also that, as in the case of margin plates, the crack occurs at the backs of the attachment angles but not at the rivet holes. The fatigue in the plate resulting in a crack seems, therefore, due to a slight but frequently repeated movement in the plating at the chock angle attachment. This suggests the necessity for securing the intermediate frames to the panting stringer and for distributing the resistance to panting stresses over a larger surface of shell plating.

By attaching the intermediate frames (unsupported by beams) to a wide stringer plate by means of substantial brackets, and by substituting double for the single chock angle attachments of the stringer to the shell plating, it is hoped that an efficient remedy has been found for this trouble.

It may be added that both in the cases of fatigue at margin plates and in those described in this paper, tensile and other tests made upon material cut from the vicinity

of the cracks do not show any abnormal conditions, but simply give the ordinary test results of good mild steel used for shipbuilding purposes. This would seem to show that the effects of fatigue are practically limited to the small portion of the material which yields under its influence.

The study of what was well termed by the late Mr. William Denny "the morbid anatomy of ships" is profitable to the shipbuilder and shipowner. It is also an essential part of the training of a ship surveyor. It is hoped that this brief description of a morbid symptom, its diagnosis and the consequent prescription, may not have unduly occupied the time of the Institution at these meetings.

DISCUSSION.

Mr. J. Foster King (Member of Council): My Lord and Gentlemen, I think the gratitude of this Institution will go out to Dr. Thearle for his paper, not for its shortness—because we would have liked a great deal more from him—but for the fact that he has opened up a page of that huge volume of practical experience which lies at his disposal. Dr. Thearle's illustration of a phase of that experience in relation to practical shipbuilding is, in a sense, a return to the earlier and, I think, healthier days when discussions of defects and their causes were more frequent than they are to-day. The diffidence which makes it difficult for naval constructors and marine engineers to describe the structural details of features which make for success seems to render them almost inarticulate in regard to things which have not been satisfactory, in spite of their great teaching value.

With reference to failures of margin plates at their connections, I scarcely think Dr. Thearle quite means that his experience of these defects has been entirely limited to that of 20 years ago, because my experience at least has shown the need for continuous development in the strength of these connections. When my Society first introduced their rule for gusset connections, the method of attachment was so much more satisfactory than the ordinary lug connections then in use, as to suggest that a final cure had been found, yet as years passed the increase in dead-weight capacity and of power in proportion to the bulk of the ship which has been the result of evolution, has gradually caused earlier experience to pass out of date, and on at least one occasion during these twenty years it has been necessary to increase the standard of attachment, while no later than this year, observation of our fleet has led my Committee to considerably augment and regulate this attachment in order to keep pace with the demands of development.

With reference to the particular phenomena to which Dr. Thearle has referred, I must confess to having no personal experience of the exact nature which he has described, but I do remember a year or two ago one of these sporadic outbreaks of trouble which characterise shipbuilding experience and which, I think, are due to the closeness with which shipbuilding practice follows the dividing line between success and failure. Quite a number of ships, within a year or two of their completion, were coming into port with leaky shell chocks in the fore peak, and the defects were at first attributed to bad work-

manship, because there is no doubt as to the difficulty of fitting short shell chocks having a big bevel and a curved form. The lugs were in each case carefully refitted and re-riveted, and as the trouble recurred it became obvious that something else was the cause. About this time it had become the fashion to somewhat reduce the standard of size for the frames in the fore peaks, and to do away with the lug attachment of the inner stringer angles, so that it was inferred that the trouble was due to panting in the frames. The probable cure appeared to be that now suggested by Dr. Thearle, and bracket plates or angles were fitted to connect the intermediate frames to the stringer plates, with the result that there was no further trouble, and these connections have been adopted as the standard of modern practice in fore peaks.

With reference to Dr. Thearle's suggestion that double shell lugs should also be fitted, I take leave to question the soundness of that deduction. It is quite true that one cannot well build the frames of a ship so that there will be no deflection between the points of support, but it is possible to so design the frames as to keep the deflection within limits which will not produce undue stress upon plating and riveting, and I venture to think that my lack of experience of disturbance in holds, of the character described by Dr. Thearle, is due to the fact that it has been the practice of the British Corporation to fit deeper framing in the fore hold than Lloyd's Register appears to consider necessary. There are two schools of practice in this matter; one makes the frame or other girder of sufficient strength, as indicated by calculation or experience, but of comparatively thick and shallow section in relation to the span, the other makes them of similar weight, but of greater strength and stiffness by the use of deeper sections. I give my adherence absolutely to the second school, because I do not think it good practice to fit any girder of so small a depth in relation to the distance between the points of support as to make it necessary to cloak the stresses at the ends by means of extra riveting, and I think Dr. Thearle's paper is more or less a confirmation of that view.

* Since I had the privilege of reading Dr. Thearle's paper, and of making the foregoing remarks, special inquiries have been addressed to individual surveyors of the Registry, in case the examination of the reports which had been made had failed to reveal all available information. As a result of the light from Dr. Thearle's paper, it appears that in some half-dozen cases the surveyors are inclined to think that the plating in the fore peak may have cracked as the result of panting strains, although hitherto this had been attributed to external causes, but so far there does not appear to have been any evidence of such disturbance in the fore hold. This in itself is a tribute to the educative value of such papers, as hitherto there was a tendency to maintain that serious panting movement could not occur without rivet disturbance.

Mr. C. E. Stromeyer (Member): My Lord and Gentlemen, Dr. Thearle's paper is certainly one of great interest, and deserving of the most careful consideration of all those who study the question of the fatigue of metals, but those who, like myself, may wish to go more deeply into this matter would like to have had more information than we have got. The information is complete in so far as that Dr. Thearle is satisfied that he has effectively counteracted the evil, but there are certain matters about which fuller information would have been welcome. The first is with regard to the weather which these ships have experienced. We hear that some ships have survived these stresses for two years, and others for 16 years, and it would naturally be very interesting to know whether the one set of ships has experienced severer weather than the other. The logs and the abstracts of logs are preserved for many years, so that the information could be obtained if an effort were made in that direction. In connection with this matter I might just point out that there are about 32,000,000 seconds in a year,

* Added after the Meeting.

and if allowances be made for the time that a vessel is in port and for calm weather, there will remain a very short period in the year during which a ship is experiencing severe weather. I should imagine that in one year the number of alternations of stress would certainly not exceed a million, and might be much below this number. This would mean that for those ships which lasted for two years, one may, for want of fuller information, assume one million alternations of stresses, and in the other cases eight million alternations may be assumed.

Based on experiments which I am completing (and which now number about 400) on the endurance of mild steel, the stresses which will cause fractures in, say, two to eight million alternations, should be respectively about 14 tons to 12 tons per square inch. The figures are for acid open hearth steel, and they suggest the second point about which we should have liked fuller information, viz., which of the ships were built of basic, and which of acid material, distinguishing, at least by letters, the various makers. If that could be done, I would like to hazard the guess that certain steel makers who adopt slightly different principles of manufacture to those of others, would not be found in this list, or, if they did appear as having supplied steel for any of the damaged ships, it would only be to those which gave way after about 16 years' service.

The third subject about which we should have liked fuller information is the actual stresses which occur near the fractured points. Lloyd's Register Committee have had before them my strain indicator since 1885, for it was explained to them before I read my paper at this Institution, it has been used by the Admiralty on the *Wolf* experiments,* and has given very valuable information, and a large number of other Admiralties are using it for determining strains in ships. I cannot see why Lloyd's Register should be content to rely on failures like the present ones when instruments—not necessarily mine—for measuring stresses in ships are available and in practical use.

Mr. W. H. Whiting (Member of Council): My Lord and Gentlemen, I entirely agree with Mr. Foster King that we are very much indebted to Dr. Thearle for resuming a very welcome practice in connection with this paper. I have never met, and I do not recollect hearing, in regard to our warships, of this particular difficulty, but I should like to make two suggestions with regard to it. The first is this: Dr. Thearle mentions on page 30, three features relating to the places in which these cracks are found. He has not, I think, noticed this point: That in four out of the seven cases it occurs just opposite the lower tier of beams, and that those are the cases in which the cracks occur most numerously. It is true that there are some other cases in which they occur on the tier above the lowest, but in those cases they are only single cracks in each case, whereas when they occur quite numerously it is always in connection with the lower tier. That leads me to suggest as a possibility that the cracks are due to the drumming action of the unsupported part of the plating below the beams, the hard edge formed by the angle bar determining the rapidly reversed local stresses which I presume can alone account for these cracks. The other suggestion, or rather question which I should like to ask Dr. Thearle, is whether he has noticed anything special with regard to the occurrence of these cracks in relation to the butts of the plates. I should like to know whether they generally occur on plates which end on the stem, or in which some special stress may have been set up in the actual working of the plate. I do not say that this is a probable cause, but it may have some bearing on the matter, and in any case the matter is so obscure and at the same time so interesting, that it is well to exhaust every possible direction of inquiry.

Mr. J. R. Jack (Member): My Lord and Gentlemen, I should like to add my thanks to those who have already gone before me, to Dr. Thearle for bringing this subject before us. To the ship-

* Trans. I.N.A., Vol. XLVII. (Part I.), page 80.

builder, dealing with a new ship, such diseases do not present themselves, and it is only by having them put before us that we can know what to do in the future. The number of cases quoted is fairly large, some 300, but when we consider that number in comparison with the enormous number of vessels coming under Lloyd's survey which have not shown any signs of this weakness at all, it does not appear to be a very big thing, and the question is, if other vessels have lasted for many years without showing any sign of this disease, why should it occur in this comparatively small number? Personally, I feel inclined to put it down to workmanship. That is a somewhat dangerous thing to say, but those of us who have to deal with these matters know that workmanship, even in the best yards, is liable to, say, variation. With the best intentions in the world we cannot be sure that every part of the work is absolutely perfect. We may get along fairly well, but the head foreman may be off ill, and the under foreman may not be quite capable of keeping things up to the standard, and something not as good as we should like it to be goes through. That being so, it appears to me that the probability of this fatigue, which is certainly due to the "working" of the plating itself, is due to slightly deficient workmanship, and this is exactly the place where one would expect it to occur; there is a little slack in the plate in all probability, and that slack "works" in the extremely varying conditions obtaining in the fore peak, and that is where the strain has always shown itself. Frames in the fore part are necessarily very much bevelled, and unless the work is very carefully done the flanges are slightly hollow. One can go to almost any dock and look at almost any ship, and find that in nine cases out of ten there are signs of this disease in those places; there is a little bit too much length of shell plating causing a bulge, which is capable of yielding under the blows that it certainly gets in that locality and, I think, if something could be done—it is a very difficult thing to put a standard of workmanship—to guard against this "slackness"; to ensure that the frames were so bevelled that any concavity shall be in the athwartship flange and not in the shell flange, and that the plates should be a little short so that they may be put on with initial tension, and this slack avoided, I think we should get out of the trouble. I do not know if the surveyors who reported these cases made any mention of that fact, but it would be interesting to know if the shell plating in way of those defects showed any amount of superfluous material. That appears to me one reason why these things might occur. Looking back at the number of vessels in which they did not occur, and which proved that the methods adopted were quite sufficient, I should be inclined to ask whether they could not be made sufficient for all other vessels as well.

Mr. S. J. P. Thearle, D.Sc. (Vice-President): My Lord and Gentlemen, I feel deeply indebted to those gentlemen who have been kind enough to speak upon this paper. My object in writing the paper was not merely to give information but to get it. I was desirous of hearing the views of this meeting upon a phenomenon which had puzzled me somewhat at first, and of which I was not certain that I had even yet found the correct solution. My object therefore was partly to do what I think is the duty of every member, namely, to communicate his experiences to his colleagues, and in order to get the experience of the meeting to assist me in determining the actual cause of the phenomenon referred to. I have first to thank Mr. Foster King for his kind reference to the paper. He seems to doubt whether I am correct in saying that I have had no recent experience of the defect formerly so common in margin plates. I assure Mr. King that during the last fifteen years no case of that kind has come under my attention. I cannot, of course, say positively that they have not happened, or that our surveyors have not seen them, but I can say that they have not been brought to my notice, and I think they would have been had they occurred. I was under the impression that the trouble was entirely remedied and might now be considered a thing of the past,

but as Mr. King says that he has come across some sporadic cases lately it simply shows that they may still be looked for. Mr. Foster King seems to think that the prime cause of the trouble to which my paper refers is deficiency of frame strength at the fore part of the vessel.

Mr. Foster King: Stiffness, not strength.

Dr. Thearle: A deficiency of frame stiffness. Mr. Foster King suggests that the remedy is, while using the same weight of material, to alter the section of the frame, by making it deeper and thinner. I admit that on paper the frame made deeper and thinner with the same weight of material gives a better calculated result, but I do not like the frame section so produced, because there is very little upon which to come and go with regard to wastage by corrosion, and because it is not laterally a stiff frame. However, I agree so far with Mr. Foster King in thinking that much may be done in the way of stiffening up vessels at the fore part. I am personally of opinion that the trouble we are discussing to-day is largely due to vessels making ballast voyages; that is to say, voyages in a light condition, and then encountering bad weather. I cannot speak with certainty upon this matter, but I have a shrewd suspicion that these troubles do not occur when a vessel is loaded down to her marks.

Mr. Stromeyer is like Oliver Twist in asking for more, and if Mr. Stromeyer will call at my office in London I will give him some little pieces of the plate that he may take away and analyse and test at his leisure, and I hope if he does so he will favour this Institution with the result of his investigation. I may tell him that although I have no absolute certainty with regard to the matter, yet I think it highly probable that all, or nearly all, the steel which has been found to fail in this way was acid open-hearth steel. Each case has been that of a vessel built by a British builder, and in this country acid open-hearth steel is for the most part used in the construction of ships. Hence I think most of the steel was made by the acid process. There might, of course, have been occasional instances in which basic open-hearth steel has failed, but I think on the whole we may describe it as acid steel.

Mr. Whiting wishes to know whether the positions of the cracks bore any suggestive relation to the butt of the plate. I have not, of course, personally looked into many of the cases of failure; I have had to leave that matter to my staff, but those I have seen show nothing suggestive of that kind. Most of the cracks are in No. 2 hold, so that they would not be near the extremities of the plates abutting on the stem, and with regard to other butts I have no suspicion that the position of the cracks in relation to the butts has any bearing on the question.

Mr. Jack thinks that thirty-three vessels bear a very small proportion to the total number surveyed by Lloyd's surveyors in the course of the two years during which these phenomena have been observed. That is true, but it is a remarkable fact that previous to two years ago we heard of none. It is a puzzle to me why we did not hear of these things earlier than two years ago, and that strengthens my suspicion that it is a ballast trim question. It is well known, I think, by most of us here that during comparatively recent years the world's carrying trade has been considerably modified by the American Tariff. Whereas some years ago—and not so many either—cargo vessels trading on the Atlantic took cargoes both ways, to-day they only take cargoes from America to this country, and go westward in ballast. I think, therefore, that this trouble may be considered to have started at the time when vessels commenced making voyages across the Atlantic in ballast, and that about the time when the oldest vessel in which such cracks have been found was building (which I think was about sixteen years ago) would be about the time when the vessels began to experience the particular stresses which when long repeated have caused fatigue and failure. I do not think Mr. Stromeyer is

justified in assuming that because the defect has only been observed recently in vessels sixteen years old that it has taken sixteen years to develop. I am inclined rather to think that these peculiar defects would have been visible at an earlier date if the attention of our surveyors had been more particularly directed to them. It was only after four or five pronounced cases had been observed that the attention of surveyors was particularly directed to seeking for them in cases where they were not so obvious, and from the time that the surveyors were asked to look out for these defects they began to discover them rather frequently, until at last in two years we have had reported no less than thirty-three cases.

Mr. Foster King: Do the surveyors never discover leaky rivets?

Dr. Thearle: Never a leaky rivet in association with this particular trouble nor has any association with the riveting been suggested, except that, of course, the riveting has caused rigidity at one place and not at another. It has been the absence of rigidity between two rigid places which seems to have produced the trouble. In other words, the fatigue has apparently resulted from blows of the sea coming repeatedly upon places which were not uniformly supported, and the point which was furthest removed from the points of support has been the one where the crack has occurred.

Mr. Jack thinks the trouble might be due to bad workmanship. That is a very serious reflection upon the supervision given to the work, and I cannot accept it. I suppose bad workmanship will sometimes occur in the best regulated shipyards, but this phenomenon has been observed in vessels built by seventeen different builders. Does Mr. Jack suggest there are seventeen builders in this country capable of doing work sufficiently bad to result in cracked shell plates? I cannot think so. I am still open to receive any suggestions which may lead to a more accurate solution of the cause of this trouble than I have been able to arrive at myself. I may, however, frankly say that this discussion has been a source of some enlightenment to myself, and consequently I am grateful to those gentlemen who have spoken upon the paper. To that extent my object in bringing the subject before you has been attained.

The Chairman (The Right Hon. Lord Inverclyde, Associate Member of Council): I am sure you will agree with me, gentlemen, that we are very much indebted to Dr. Thearle for bringing this valuable paper before us, and to those gentlemen who have taken part in the most interesting discussion upon it. In your name I express a vote of thanks to Dr. Thearle for bringing this question before us in his paper.

EFFECT OF FORM AND SIZE ON THE RESISTANCE OF SHIPS.

By G. S. Baker, Esq., late R.C.N.C., Member, and J. L. Kent, Esq., Associate-Member.

[Read at the Summer Meetings of the Fifty-Fourth Session of the Institution of Naval Architects, June 25, 1913; the Right Hon. Lord Inverclyde, Associate Member of Council, in the Chair.]

Introductory.

The paper deals with the variation of resistance of ships with speed and with fulness of form.

In § 1 to § 3 is given a general account of the work of the earlier writers on this problem.

§ 4 to § 6. The pressure disturbances created in the water are considered by the use of Taylor's stream-line methods.

Curves of area are used for the stream lines, these giving a more correct measure of the disturbance than the use of level lines or horizontal sections of the ship. This course is adopted in view of the fact that the area curve is the controlling wave-making feature and the water does not flow along the level lines. The effect of sharp ends and restricted channels upon the stream-line flow is also worked out.

§ 7 to § 8. The meaning and the effect of these pressure disturbances are then considered, and it is shown that the characteristic humps in the stream-line pressure curves vary mainly with two factors—length and prismatic coefficient.

§ 9 to § 10. The results of experiments with models are then given and a new law for the speeds at which transverse wave-making occurs is given, this law being in general agreement with the theoretical considerations of the previous sections. The special case of high-speed vessels is discussed.

§ 11. A peculiarity of the pressure curve of the entrance is considered and is shown to produce a hump in the resistance curves of full-ended ships.

§ 12 to § 13. The effect of long full bodies upon the coefficient of frictional resistance of the form is considered both from the theoretical and experimental sides. The results of certain comparative experiments with models in air and water are given and show the effect that theory leads one to expect.

§ 14. The formula is applied to William Froude's experiments, and a method is given for obtaining the maximum economic length of parallel body to be associated with a fixed entrance and run.

Historical.

§ 1. Many attempts have been made to formulate laws for the resistance of ships or the power to drive them. Almost invariably these have been framed using the principal dimensions, with varying constants in a rigid formula, and they generally purport to give quantitative values for the resistance or power at all speeds. The authors in bringing this paper forward do not attempt to cover such a large field as this. Their object is to draw attention to the methodical manner in which certain features in the ship affect its resistance, the relative importance of these features, and the limitations which they impose upon the possible economical speed of the ship. No attempt is made at present to give a quantitative measure of ship resistance.

§ 2. Scott Russell may be considered as amongst the first to deal with this subject from the correct point of view. The underlying principles of his "wave-line system" of construction are given in three papers read before the Institution of Naval Architects in 1860-61.* It is not necessary to examine the principles and arguments on which this system was based, but we may give the two chief rules which Mr. Russell formulated for obtaining the best proportions for minimum resistance :—

(1) The bow lines should be versine curves and the stern lines should be cycloids.

(2) The lengths of entrance and run† necessary for any speed were those in the following table, there being nothing gained by using greater lengths and something lost by the use of shorter lengths.

It was soon recognised that the entrances given above were too great, and Professor Rankine advised the snubbing away of the extreme end of the bow.

Although practical experience and model experiments have since shown that the versine curve of areas is good from a resistance point of view at moderately high speeds, it is known to be too fine at the ends for either really high or comparatively low speeds. This fact and the low prismatic coefficient which a versine curve involves renders its use the exception rather than the rule.

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* The system was first announced in a paper read before the British Association in 1834, and the *Wave* was built on this system in 1835.

† By entrance and run here is meant that portion of a ship at the bow and stern, respectively, over which the shape is changing. That portion of the body near amidships of which the transverse section remains the same for any length is called parallel body, and this separates the entrance from the run.

§ 3. There are now many records of experiments on ship model resistance, given in papers by such able experimenters as W. Froude, R. E. Froude, D. W. Taylor, &c. Mr. R. E. Froude has also expounded the general principles of ship resistance.

TABLE I.

LENGTHS OF ENTRANCE AND RUN FOR GIVEN SPEEDS.

Statute Miles per hour.	Length of Entrance in feet.	Length of Run in feet.
2	1·68	1·2
4	6·72	4·8
8	26·88	19·2
12	60·48	43·2
16	107·52	76·8
18	136·08	97·2
20	168·00	120·0

Of these papers the two most important and most akin to this paper are those by W. Froude* in the Transactions for 1877, and R. E. Froude† in 1881. The former gives the results of experiments with models whose sole difference was due to varying lengths of parallel body inserted between the entrance and run. The paper showed clearly, how the residuary resistance varied with the length proportions of the model. The latter paper gives the general principles of the resistance of water to the motion of a ship.

As a rule the problem set the designer is not so much concerned with the effect of elongation of a ship, but the effect of adding displacement on given beam, length and draught, either as parallel body, or in entrance and run.

The authors have considered the problem from this point of view more than any other and hope to have shed some little light on it.

THEORETICAL CONSIDERATIONS.

§ 4. The problem may be approached from the standpoint of stream lines in a perfect fluid, and on certain assumptions such stream lines may be obtained. The process adopted by the authors in doing this is that given by Mr. Taylor‡ in 1894 for flow in two

* Trans. I.N.A., Vol. XVIII., p. 77. † Trans. I.N.A., Vol. XXII., p. 220.
‡ Trans. I.N.A., Vol. XXXV., p. 385.

dimensions. This consists of combining graphically a line of sources and sinks of varying magnitude with a parallel stream, the distribution of the magnitude of the sources and sinks being altered to produce the differently shaped stream forms.

The method is illustrated in detail in Appendix I., in which are given the various steps necessary to obtain the stream forms and their pressure curve.

The two dimensional case has been used for two reasons :—

(1) It saves a large amount of laborious work.

(2) By using the ship's curve of areas for the stream form, instead of a level line of the ship, we get a better indication of the mean pressure variations along the ship which the area curve represents, both with change in prismatic coefficient and longitudinal distribution of displacement on the same prismatic coefficient.

Although the assumption made in condition (2) above may not be rigidly true, it affords a better criterion of pressure variation than that obtained by the use of level lines. The stream lines on the model surface obtained experimentally by Mr. Taylor show that the motion is seldom or never in the horizontal plane. Experiments made with a high-speed model at the William Froude Tank have shown much the same thing. Moreover, there is ample evidence that transverse wave making is far more dependent upon the curve of areas than upon the shape of individual level lines, and this is particularly true with that phase of the question we are mainly concerned with here, viz., the interference of the bow and stern systems with one another.

§ 5. Altogether eleven cases have been examined in detail, and Fig. 1 (Plate III.) shows six* of the stream forms obtained and the distribution of pressure along them, and their general dimensions are given in Table II.

TABLE II.

General Dimensions of Stream Forms.

Form.	A	B	C	D	E	F	G	H
Length units	188	192	188	192	192	192	192	192
Midship section	25·6	25·6	25·6	25·6	25·6	25·6	25·6	25·6
Prismatic coefficient ...	·503	·554	·648	·720	·743	·751	·757	·776

* Three others had sharp angular ends, and are discussed in § 6, the remaining two are omitted from the diagrams for the sake of clearness.

It will be seen that these have the common characteristic of all such forms: pressure humps near the ends and decreased pressure at the middle. It will also be seen :—

(*a*) That the humps are generally more emphatic the fuller the form, and occur nearer the ends up to a certain prismatic, beyond which increase in fulness has little effect.

(*b*) That the forms with the larger prismatics have a curious depression in the pressure curve occurring between the maximum pressure and the mid-length of the body.

Moreover, this depression approaches the pressure hump more closely as the prismatic coefficient is increased, as is shown in columns 3 and 4 of Table III., and in consequence of this, the area of negative pressure in the limits of the length of the form is considerably greater with the larger prismatics.

TABLE III.

PRESSURE CURVES.

Form.	Prismatic coefficient.	Longitudinal position of maxima and minima as % of half-length from amidships (§ 8).		Area of negative pressure curve from amidships to position of zero ordinate.	Mean ordinate of pressure curve over the total length of stream form (§ 8 and 12).
		Positive hump.	Negative hollow.		
A	·503	78	0	·193	·128
B	·554	80	0	·235	·160
C	·648	84	35	·246	·185
D	·720	89	53	·246	·193
E	·743	90·5	56·5	·282	·212
F	·751	89	58	·240	·180
G	·757	90·5	63	·297	·249
H	·776	89·5	55	·283	·225

But the longitudinal distribution of this negative pressure varies systematically with the change in prismatic coefficient. With the finer forms the negative pressure increases continuously towards the midship section, but with the fuller forms there is a

distinct waviness in the curve. This waviness is most emphatic in the Form H, which has its largest ordinate at about one-third of its length from amidship, but it is almost as noticeable in Form F, which has 21 per cent. of its midship length parallel.

§ 6. If the stream form is given an angular termination, the pressure hump has its maximum ordinate at the end of the vessel, and the larger the angle of entrance of the form the more peaked does the curve become at this point. The more marked this peak becomes the greater is the area of the pressure curve on the positive side of the peak, *i.e.*, to the right in Fig. 1, and the more sudden is the drop on the negative side—the negative pressure extending over a greater length of the form. These changes are much as we should expect from the previous work, since increase in angle of entrance is usually accompanied by extra fulness, *i.e.*, by increase in prismatic coefficient.

The stream lines and pressure curves for such a form are given in Fig. 5 (Plate IV.), which is also interesting in that it shows the effect of the presence of boundary walls on the stream-line motion. The distance between these walls is equal to the length of the model, which is equal to six beams. It will be seen that the pressure changes amidships are approximately 30 per cent. greater with these restrictions than without them. Herein lies the explanation of the growth of resistance when a ship runs into either shallow water or a narrow channel. Due to the greater changes in stream-line flow there is a substantial increase in the velocity of the streams in contact with the form, resulting in an increase in the skin friction, and under unfavourable circumstances (for propulsion) the wave making is also increased. The method of obtaining these stream lines with the boundary walls is new so far as the authors know, and is given in Appendix II.

Meaning of the Curves.

§ 7. The effect of certain types of pressure disturbances upon the surface of water is given in text-books on hydrodynamics, and in the papers of Lord Kelvin and Professor Havelock. Of these, that which bears most resemblance to our present problem is the travelling point of pressure, considered by the two authors already mentioned.

This produces a system of waves of the usual transverse and divergent character, the whole being included between lines inclined at 19° 28′ to the line of motion, the amplitude of the waves decreasing as the square root of their distance from the origin.

The effect produced by two of Lord Kelvin's pressure points correctly spaced must be somewhat the same as that of the two humps in the pressure curves of Fig. 1. The differences between the two cases are mainly :—

(*a*) The pressure is somewhat distributed in the ship, and not in the case of the pressure points.

(*b*) The pressure acts upon water in motion around the ship, and upon still water in the case of the pressure points.

(*c*) Any wave motion set up causes a change in the pressure system around the ship.

(*d*) There is a wide band of negative pressure between the two humps in the ship which is tending to produce its own set of transverse waves, and which has no counterpart in the case of the pressure points.

For these reasons, although we can expect some likeness to exist between the two cases, the differences are too great to allow of any comparison of results except on the broadest lines. It is enough, however, for our purpose, to be able to state that such pressure disturbances as we have shown exist when a ship is in motion will produce waves which will vary, more or less, in accordance with the results given in the papers by the above authors. The character of the resultant system of waves formed by both bow and stern will vary with the relation between the speed and the distribution of the pressure systems. The bow pressure hump generates its own system of divergent and transverse waves, which are very similar * to those produced by a single point disturbance. But this hump is followed by a hollow and a second hump, which are also tending to create other systems of waves of the same period. When these pressure effects are so situated that they tend to produce waves having their crests and hollows superposed upon each other, bad wave-making and high resistance will result. Moreover, the undulation of the negative pressure near amidships will affect the wave-making, and according to the speed so this may add to or detract from the total wave-making effect.

The examination of the pressure curves has shown that the relative positions of these humps and hollows are dependent upon two things, viz., the length of the stream form, and its prismatic coefficient, and we are led to the conclusion that the wave-making speeds of any ship depend directly upon these two factors.

The above remarks apply more particularly to transverse wave-making—in which interference of one system with another is of great importance, owing to the large amount of energy involved in them at the usual ship speeds. The diverging waves depend almost entirely upon the comparatively local bow and stern humps, and it is seen that these will generally increase with the prismatic coefficient.

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* The angle of divergence, in the ship, depends upon the speed and form of the ship, decreasing with increase of speed and to a certain extent with fineness of ends. In the case of the single point of pressure this angle is constant.

§ 8. Since the resistance of a form depends partly upon skin friction, and this varies approximately with the square of the velocity, it will be seen that the extension of the region of negative pressure is equivalent to an increase of rubbing velocity of the stream lines against the form. The area of this negative pressure curve is a direct measure of this effect for the portion over which it extends, and column 5, Table III., shows these values for the various stream forms. As a partial counterbalance to this, there is a decrease in velocity at the ends, and column 6 gives the mean change in $(\text{velocity})^2$ taken over the whole stream form. It is difficult to say how much of this increase in rubbing velocity would be realised as additional skin friction in a ship. The frictional resistance is due to a belt of water and not to the film which is in contact with the ship, and as the variation of velocity in the stream lines is less the farther these lines are removed from the form, there will be a negative correction to be made for this. In one case which has been fully examined, this correction taken on a belt which erred on the larger side did not amount to more than 20 per cent. Moreover, it must be remembered that where the stream velocity is greatest, there the area of skin in contact with the water is greatest, and this requires a positive correction which may well balance that already considered.

Experiment Results.

§ 9. In order to test these results the resistance experiments with some 215 models have been examined. These vessels include broad and narrow, shallow and deep, fast and slow vessels, and cover a very extensive range of prismatic coefficients. The resistances of the models tested at the experiment tank are tabulated in the Froude constant form, which for present purposes may be regarded as $\frac{\text{resistance}}{(\text{velocity})^2}$. Typical constant curves are shown in Figs. 3 and 4 (Plate III.), and it will be seen that these are generally constant in ordinate value, but have humps of increasing value as the speed increases. These humps are succeeded in some cases by hollows and in others by a general flatness of the curve. As pointed out by Mr. Froude many years ago, when any two features of the ship or model tend to create waves having crests and hollows in the same fore and aft position, the superposition of these waves means abnormal wave-making, therefore abnormal resistance, which is shown by these humps. It will be seen on reference to Fig. 3 that there is a particular speed at which the hump is extremely large. With the exception of only one class of ship (which is dealt with later on) the change-over from this abnormal hump to the succeeding hollow has been found to occur at the natural speed of waves whose length is equal to $(P \times L)$ where

P is the prismatic coefficient and
L is the length of the model in feet.

This speed is given by the formula:—

$$V = 1{\cdot}34\ \sqrt{PL}$$

V being the velocity in knots.

The speed at which the other and smaller humps occur can be found by substituting $\frac{L}{n}$ for L, n being an integer which takes account of the number of wave lengths between the two wave-making systems of the model. This invariably gives the correct speed for the flat portion of the hump marked A in Fig. 2, and it also gives the second and third of these points at lower speeds with fair accuracy when the ship is full enough to make them noticeable. At low speeds for the form the formula is not so trustworthy, as the pressure distribution changes a little with speed, but as the wave-making is then of little importance, this does not detract from the usefulness of the formula.

The curves of Fig. 2 (Plate III.) show the results of resistance experiments on five models. They are plotted to abscissæ values

$$\text{Ⓟ} = \frac{\frac{6080}{3600} \times V}{\sqrt{\frac{g\,P\,L}{2\,\pi}}} = {\cdot}746\,\frac{V}{\sqrt{P\,L}}$$

the ordinate value being equal to $2{,}983\ \frac{R}{(\text{displacement})^{\frac{2}{3}}\,(\text{velocity})^2}$. The critical speeds of all the models correspond to—

$$\text{Ⓟ} = 1, \frac{1}{\sqrt{2}}, \quad \frac{1}{\sqrt{3}}, \quad \&c.,$$

and there are flats or hollows in the resistance curves at these abscissæ values.

§ 10. The exceptional class which has been mentioned above includes all vessels forced to a high speed for the form, such as destroyers, steam pinnaces, &c. These vessels show the hump for $n = 1$, *i.e.*, Ⓟ = 1, but owing to the fine lines usually adopted, this hump is not of excessive magnitude, and the vessels may be forced to speeds much in excess of the velocity corresponding to it. As the speed is increased above Ⓟ = 1, the value of Ⓒ (or $\frac{\text{resistance}}{(\text{velocity})^2}$) at first remains fairly constant, and then increases very considerably. Simultaneous with this increase in resistance is a marked lifting of the bow and sinking of the stern. With further increase in speed, the value of Ⓒ decreases, and the trim settles down to a fairly constant figure. During these changes the vessel first sinks, and after the maximum value of Ⓒ has been reached, it very slowly rises in space with increase in speed. One other change takes place during this phase. At Ⓟ = 1 the divergent waves are of the ordinary stream-line character; but at Ⓟ = 1·5 the surface water reaching the bow is thrown to one side along the

diverging wave crest, and the energy involved in this is a dead loss, irrespective of the wave-making features.

This large "hump" in all the distroyers whose results we have been able to examine occurs at a speed given approximately by :—

$$\text{(P)} = \sqrt{2\tfrac{1}{4}} = 1{\cdot}50$$

It represents that phase of wave-making when both entrance and run tend to form coincident systems at the stern. At higher speeds than this, conditions improve so far as interference is concerned, and the "planing" action of the stern also tends that way. A model tried at speeds ranging from 30 to 50 knots for a vessel of 250 ft., showed less transverse wave-making as the speed was increased to the upper limit. This was particularly noticeable to an observer stationed at the beach at the end of the tank waterway. Unfortunately the divergent waves increased continuously, and the percentage of resistance due to wave-making remained fairly constant at all the top speeds. The resistance records in this case were only taken up to 45 knots for the ship, but in another case, the records were taken to 50 knots, and showed a continuous decrease in wave-making at the higher speeds.

§ 11. There is one other class of hump which occurs in the resistance of a ship, and to which attention should be drawn. Except in the case of vessels of full form, this hump is not of much importance, but it may easily become so if it agrees with the hump given by $n = 2$ or 3 in the formula of § 9. Fig. 3 (Plate III.) shows the "constant" curves for a series of models, and it will be seen that there is a hump which travels along the curves to higher speeds as the relative length of entrance to run is increased. A similar set of curves for fuller forms is given in Fig. 4, which show the same characteristic, but not quite so clearly, as the humps are due to the total wave-making, and include any resistance caused by the bow and stern combination already dealt with. The position of this hump evidently depends upon the length of entrance of the model. Its magnitude is practically independent of this, and varies as one would expect it to vary with the speed at which it may occur. This speed is partially dependent upon the form of entrance varying a little according to the fulness or hollowness of the ends. Approximately it is given by the formula :—

$$V = 1{\cdot}095\ \sqrt{L_E}$$

V being the speed in knots and L_E the length of entrance of the form in feet. This is the speed of a trochoidal wave whose half length is $\frac{1}{3}\ L_E$. So that it is reasonable to suppose that the hump is caused by interference either of two pressure humps of phase interval equal to $\frac{2}{3}\ L_E$, or of a pressure hump followed by a depression at an interval of $\frac{1}{3}\ L_E$. All the full forms whose pressure distribution has been considered have such a

depression following the hump near the bow, and it would appear that this is the cause of the hump in the resistance curve. It will be seen that when the entrance is long enough, the speed for the entrance wave hump may coincide with that speed which gives a bad resultant transverse wave system. These two formulæ have been of great use to us at the Experiment Tank, and to just the same extent they are useful to any ship designer. A study of Fig. 2 (Plate III.) will give the (P) values which are likely to be troublesome at different prismatic values. Once these are obtained for any parent form, the position of the humps for any modified form can be predicted with considerable accuracy.

Friction.

§ 12. A few of the model results have been examined to see whether these show any tendency to higher skin friction values, when the prismatic coefficient is increased by the use of parallel body. Only the lowest speeds can be used for this purpose, in order to get away from possible wave resistance, and even then a small percentage of the resistance may be due to the divergent waves.

Neglecting this, however, and considering the whole resistance as frictional, it has been found that for models having 50, 30, and 10 per cent. parallel body the value of the constant f in the formula—

$$R = f A V^{1\cdot825}$$

is ·21, ·135, and ·10 greater than the corresponding f values for a plank of the same length as the model considered. The above are average figures, as the results vary somewhat amongst themselves according to the form of the model; but they show the same general variation as the figures in column 6, Table III., would lead one to expect.

§ 13. In order to obtain a reliable figure of the effect of form upon the skin resistance at low speeds, advantage has been taken of the formula given by Dr. Stanton in his paper read before this Institution in 1912.* This formula enables one to predict the resistance of a form in one fluid from results obtained in another. Experiments made in water in the usual way with a 16-foot model were compared with experiments in air made with a 3-foot model, which was symmetrical in shape, each half being the same as the underwater body of the 16-foot model.

The experiments in air were carried to a speed of 50 ft. sec., and corresponded to a velocity of 46 ft. per minute for the 16-foot model in water. The resistance predicted in this way was found to be 9 per cent. higher than the skin friction calculated from Froude's constants, and to vary approximately as the 1·85 power of the speed. The results in water were about 10 per cent. higher than those obtained by Froude's constants, but fell away badly below a speed of 90 ft. per minute; and the above 10 per cent. holds for speeds somewhat above this when possibly a little divergent wave-

* Trans. I.N.A., Vol. LIV., page 48.

making is present. This form had very fine lines, so that eddy-making was reduced to a minimum, the prismatic coefficient being ·60. These results lend considerable support to the suggestion already made, that owing to the stream-line motion the skin friction of the form is greater than that obtained by the use of Froude's coefficients, this increase depending upon the fulness of the form and being of the order mentioned above.

Economic Parallel Body.

§ 14. The practical value of the formula given in the paper, as a clue to the amount of parallel body which can be usefully associated with a given entrance and run, is shown in the following analysis of W. Froude's* series of experiments with varying parallel body.

The formula

$$\textcircled{P} = \cdot 746 \frac{V}{\sqrt{P\,L}}$$

may be written

$$L_P = \frac{(\cdot 746\ V)^2}{\textcircled{P}^2} - \left(P_e\, l_e + P_r\, l_r \right) \qquad (1)$$

where

P_e = prismatic of entrance. l_e = length of entrance.
P_r = prismatic of run. l_r = length of run.
L_p = length of parallel.

By its use in this form L_p can be found for any given velocity and $\textcircled{P}$ value. Determine the value of L_p for each velocity for which the resistance curves are given (1) when $\textcircled{P} = 1$; (2) when $\textcircled{P} = \frac{1}{\sqrt{2}}$; (3) when $\textcircled{P} = \frac{1}{\sqrt{3}}$, and so on.

In Fig. 8 (Plate IV.) square up from the abscissæ corresponding to these L_p values to the proper resistance curves. Draw the curves $\textcircled{P}_1$, $\textcircled{P}_2$, $\textcircled{P}_3$, &c., through the spots so obtained, all the spots on each curve corresponding to the same $\textcircled{P}$ values.

Considering the resistance curve for one velocity (say, 14·43 knots), it will be seen that the spots corresponding to—

$$\textcircled{P} = 1,\ \frac{1}{\sqrt{2}},\ \frac{1}{\sqrt{3}},$$

for this velocity all lie approximately on a mean resistance-line drawn parallel to the base. Such a straight line must be the measure of the wave resistance when the cosine term in the equation

$$R_a \propto \left(H_1^2 + H_2^2 + 2\,k\,H_1\,H_2 \cos \frac{g\,m\,l}{2\,\pi} \right)$$

* Trans. I.N.A., 1877, Vol. XVIII., page 77.

is zero. This is in accord with the statement in the body of the paper, viz., that the velocity given by the formula—

$$\textcircled{P} = \cdot 746 \frac{V}{\sqrt{PL}}$$

is approximately the velocity at which the total transverse wave-making resistance is that due to the entrance alone, added to that due to the run alone.

From the position of the points p_1, p_2, p_3, &c., on this resistance curve, it is seen that when the speed and length of parallel body of a ship are connected by formula (1), then a small increase in the length of parallel body (associated with the same entrance and run) will be bad from a resistance point of view at this velocity, whilst a small decrease will be advantageous.

The hollows of this resistance curve occur approximately a quarter of the distance between two successive spots, such as p_1 and p_2, to the right of the L_P value given by the formula. It is therefore possible to write down approximate lengths of parallel body at which the resistance is a minimum. Such lengths are given by—

$$L'_P = L_{P1} - \left(\frac{L_{P2} - L_{P1}}{4}\right) = \frac{3}{4}\left(\cdot 746\, V\right)^2 - \left(P_e l_e + P_r l_r\right),$$

$$L_P = L_{P1} \left(\frac{L_{P2} - L_{P1}}{4}\right) = \frac{7}{4}\left(\cdot 746\, V\right)^2 - \left(P_e l_e + P_r l_r\right),$$

and so on.

In other words, the most economical speeds for a ship will be given *approximately* by—

$$V = \sqrt{\frac{4}{3}\left(\frac{1}{\cdot 746}\right)^2 PL} = 1{\cdot}546\sqrt{PL} \text{ or } \textcircled{P} = 1{\cdot}153 = \sqrt{\frac{4}{3}}$$

$$= \sqrt{\frac{4}{7}\left(\frac{1}{\cdot 746}\right)^2 PL} = 1{\cdot}014\sqrt{PL} \text{ or } \textcircled{P} = \quad 756 = \sqrt{\frac{4}{7}}$$

$$= \sqrt{\frac{4}{11}\left(\frac{1}{\cdot 746}\right)^2 PL} = \quad \cdot 807\sqrt{PL} \text{ or } \textcircled{P} = \quad \cdot 602 = \sqrt{\frac{4}{11}},$$

and so on.

Any other form can be treated in exactly the same way, and bad transverse wave-making due to interference can be avoided by the proper choice of length and prismatic coefficient.

APPENDIX I.

D. W. Taylor's Method of Obtaining the Stream Lines and Pressure Curves for the Flow in Two Dimensions Past Certain Stream Forms in Unlimited Fluid.

The method adopted is to combine an indefinitely large number of sources and sinks of small strengths, extending at equal intervals for a definite distance along a straight line, with a uniform stream parallel to the source-sink line.

Suppose the ordinates of the curve C C C C (Fig. 6) are proportional to the strengths of sources and sinks situated along the axis $X_1 O X_2$; the sources being indicated by positive ordinates and the sinks by negative ordinates. To find the stream function at any point A due to these sources and sinks. Let $k s$ denote the strength of the source at some point B, so that s is equal to B D and θ is the angle A B X_2 in circular measure.

Then the strength of the source at B $= k s d x$, and the stream function at A due to sources and sinks along

$$X_1 O X_2 = \int k s \theta d x = k \int s \theta d x = k S \qquad (1)$$

the integration extending over the length of the sink-source curve.

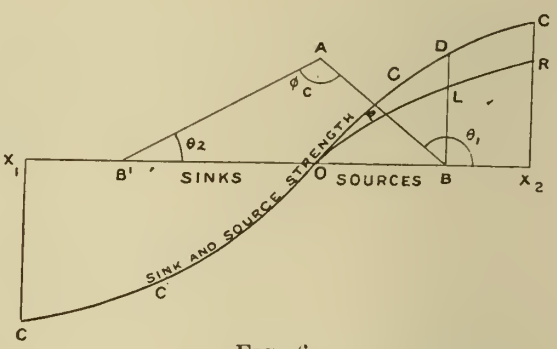


FIG. 6.

This is done graphically, a symmetrical source-sink curve being adopted to lessen the work. For the equal source and sink at B and B', the θ values at A are A B X_2 and A B' X_2 (viz.: θ_1 and θ_2).

At B set up B L = B D $(\theta_1 - \theta_2)$ = B D $\angle$B' A B = B D ϕ, say.

By carrying out this operation for a sufficient number of points along $X_1 O X_2$, the curve O P L R is obtained. The area of this curve gives the term kS, in the equation (1), for the point A, if a suitable number of ordinates along the line $X_1 O X_2$ are chosen.

In this way the stream function can be obtained for any number of spots, and enough information is obtained to draw a diagram of stream functions such as that shown in Fig. 7 (Plate IV.), which affords information for the determination of stream lines corresponding to the sink-source line.

To combine this with a uniform parallel stream in the direction from sources to sinks, the stream function at any point due to the parallel stream alone $= \psi = - v_0 y$, where v_0 = the undisturbed velocity of the stream and y the ordinate value of the point considered from the line $X_1 O X_2$.

The stream function due to the sources and sinks given in Fig. 7 is $\psi_2 = k$ S, and the compound stream function $\psi = \psi_1 + \psi_2 = k$ S $- v_0 y$.

For the stream form and for the axis X_1 O X_2 outside the stream form $\psi = 0 = k\,S - v_0\,y$.

In Fig. 7, $k\,S - v_0\,y = 0$ is the equation of a straight line through the origin. This line having one point fixed (viz., origin) can be drawn through any other chosen point.

Draw O M through the origin to the point M on the curve of "S" at station O for which y is the half width of the stream form required. The ordinates of the points of intersection of O M with each of the "S" curves are the same as the ordinates of the stream form at the same stations. The stream lines outside of this stream form are $y = \frac{k\,S}{v_0} - \frac{\psi}{v_0}$, which is the equation of a straight line in Fig. 7, parallel to O M and cutting the axis of y at a point $= -\frac{\psi}{v_0}$. Hence by drawing a number of lines parallel to O M in Fig. 7, the stream lines may be plotted in the same manner as the stream form. Such a number of stream lines are plotted in Fig. 5, which shows one quadrant, the portions of the curve in the other three quadrants being obtained by symmetry.

To Obtain the Pressure along any Stream Line.

Denote the normal pressure by p_0 and normal velocity by v_0, and w = weight per unit volume of fluid. Then along any stream line—

$$\frac{p}{w} + \frac{v^2}{2\,g} = \text{constant} = \frac{p_0}{w} + \frac{v_0^2}{2\,g}$$

whence

$$\frac{p - p_0}{w} = \frac{(v_0^2 - v^2)}{2\,g}$$

$\frac{p - p_0}{w}$ = difference between the normal pressure and that pressure where the velocity is v, expressed in "head" of fluid.

Let the velocity parallel to axis X_1 O $X_2 = v_x$.

Then actual velocity along the stream line is $v = v_x$ (secant of inclination of stream line).

Now v_x is proportional to the rate of variation of stream function ψ perpendicular to the axis X_1 O X_2.

$$i.e., \text{ to } \frac{d\,\psi}{d\,y} = k\,\frac{d\,s}{d\,y}$$

and $\frac{d\,s}{d\,y}$ is proportional to the tangent of the inclination to the axis of y of the curve "S" in Fig. 7.

The value of $\frac{d\,s}{d\,y}$ when plotted for a number of points gives the information required to obtain v_x, and by measuring the inclination β of the stream line at any point on Fig. 5, v_x sec β, *i.e.*, V for that point is known.

In this way $\frac{p - p_0}{w}$ is determined at any point. and the pressure curve for any stream line can be drawn.

APPENDIX II.

A Method of Obtaining the Stream Lines and Pressure Curves for the Flow in Two Dimensions Past Certain Stream Forms in a Restricted Channel.

The method adopted is similar to that in Appendix I., the difference being that, whereas in unlimited fluid the stream function for a source or sink is $\psi = \tan^{-1} \frac{y}{x}$, for a restricted channel of breadth b it becomes

$$\psi = \tan^{-1} \left\{ \frac{\tanh \frac{\pi x}{b}}{\tan \frac{\pi y}{b}} \right\}$$

In the place of measuring $\frac{y}{x}$ in the previous work, it is necessary to calculate $\tanh \frac{\pi x}{b}$ for every abscissa value x, and $\tan \frac{\pi y}{b}$ for every ordinate value y, and the ratio $\left\{ \frac{\tanh \frac{\pi x}{b}}{\tan \frac{\pi y}{b}} \right\}$ is used to obtain the angles θ.

The remainder of the work is exactly similar to that given in Appendix I.

Table I. gives the θ values for a number of abscissæ and ordinate values in a channel of breadth equal to length of source-sink line.

Table II. shows a typical table for deducing the stream functions for the sources and sinks, and Table III. for obtaining the pressure along one of the stream lines.

Diagrams 7 and 5 (Plate IV.) show the chart of stream functions and the stream form obtained from it.

TABLE I.—VALUE OF θ.

	ABSCISSÆ.									
Ordinates.	100	90	80	70	60	50	40	30	20	10
0	−1·571	−1·571	−1·571	−1·571	−1·571	−1·571	−1·571	−1·571	−1·571	−1·571
5	−1·487	−1·485	−1·481	−1·475	−1·467	−1·455	−1·436	−1·401	−1·347	−1·139
10	−1·400	−1·395	−1·388	−1·375	−1·351	−1·334	−1·296	−1·231	−1·105	− ·810
15	−1·313	−1·305	−1·295	−1·277	−1·253	−1·218	−1·164	−1·075	− ·915	− ·602
20	−1·228	−1·217	−1·206	−1·182	−1·152	−1·109	−1·043	− ·938	− ·765	− ·469
30	−1·061	−1·047	−1·031	−1·001	− ·963	− ·909	− ·832	− ·718	− ·551	− ·314
40	− ·896	− ·880	− ·863	− ·829	− ·787	− ·731	− ·654	− ·548	− ·405	− ·223
50	− ·738	− ·722	− ·704	− ·671	− ·631	− ·578	− ·508	− ·418	− ·302	− ·163
100	0	0	0	0	0	0	0	0	0	0

	ABSCISSÆ.									
Ordinates.	0	−10	−20	−30	−40	−50	−60	−70	−80	−90
0	0	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571
5	0	1·139	1·347	1·401	1·436	1·455	1·467	1·475	1·481	1·485
10	0	1·810	1·105	1·231	1·296	1·334	1·351	1·375	1·388	1·395
15	0	·602	·915	1·075	1·164	1·218	1·253	1·277	1·295	1·305
20	0	·469	·765	·938	1·043	1·109	1·152	1·182	1·206	1·217
30	0	·314	·551	·718	·832	·909	·963	1·001	1·031	1·047
40	0	·223	·405	·548	·654	·731	·787	·829	·863	·880
50	0	·163	·302	·418	·508	·578	·631	·671	·704	·722
100	0	0	0	0	0	0	0	0	0	0

TABLE I.—*continued.*

	ABSCISSÆ.									
Ordinates.	−100	−110	−120	−130	−140	−150	−160	−170	−180	−190
0	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571
5	1·487	1·490	1·491	1·492	1·493	1·493	1·494	1·494	1·494	1·494
10	1·400	1·404	1·407	1·409	1·411	1·412	1·413	1·413	1·413	1·413
15	1·313	1·319	1·323	1·327	1·329	1·331	1·333	1·333	1·333	1·333
20	1·228	1·236	1·241	1·245	1·248	1·251	1·253	1·254	1·254	1·254
30	1·061	1·072	1·080	1·086	1·090	1·094	1·095	1·097	1·097	1·097
40	·896	·908	·918	·924	·929	·933	·936	·937	·938	·938
50	·738	·750	·760	·767	·772	·776	·779	·780	·781	·781
100	0	0	0	0	0	0	0	0	0	0

	ABSCISSÆ.									
Ordinates.	−200	−210	−220	−230	−240	−250	−260	−280	−300	−400
0	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571
5	1·494	1·494	1·494	1·494	1·494	1·494	1·495	1·495	1·495	1·495
10	1·414	1·414	1·414	1·414	1·414	1·414	1·414	1·414	1·415	1·415
15	1·334	1·334	1·334	1·334	1·334	1·334	1·334	1·334	1·334	1·335
20	1·255	1·255	1·255	1·255	1·255	1·255	1·255	1·255	1·255	1·256
30	1·097	1·098	1·098	1·098	1·098	1·099	1·099	1·099	1·100	1·101
40	·939	·939	·939	·940	·940	·940	·940	·941	·941	·942
50	·781	·782	·782	·782	·782	·783	·783	·783	·784	·785
100	0	0	0	0	0	0	0	0	0	0

TABLE III.—STR

1. Stations ..	Station	0	30	50	60
2. Abscissæ of Stations	$\frac{ds}{dy}$	2·205	1·970	1·565	1·295
	$k\frac{ds}{dy} = \cdot10614\,v_0$ ..	$\cdot2340\,v_0$	$\cdot2091\,v_0$	$\cdot1661\,v_0$	$\cdot1374\,v_0$
3. Sink and Sou	v_0	v_0	v_0	v_0	v_0
	v_x	$1\cdot2340\,v_0$	$1\cdot2091\,v_0$	$1\cdot1661\,v_0$	$1\cdot1374\,v_0$
4. θ For Sourc	Sec θ	1·0000	1·0022	1·0065	1·0083
5. θ For Sinks					
6. Values of ϕ	Total $v = v_x$ Sec θ ..	$1\cdot2340\,v_0$	$1\cdot2118\,v_0$	$1\cdot1738\,v_0$	$1\cdot1466\,v_0$
7. Ordt. for ψ	v^2	$1\cdot5222v_0^2$	$1\cdot4681v_0^2$	$1\cdot3770v_0^2$	$1\cdot3145v_0$
4. θ For Sourc	v_0^2	v_0^2	v_0^2	v_0^2	v_0^2
5. θ For Sinks					
6. Values of ϕ	$v_0^2 - v^2$	$-\cdot5222\,v_0^2$	$-\cdot4681\,v_0^2$	$-\cdot3770\,v_0^2$	$-\cdot3145\,v_0$
7. Ordt. for ψ	Pressure from normal	$-\cdot5222\frac{v_0^2}{2g}$	$-\cdot4681\frac{v_0^2}{2g}$	$-\cdot3770\frac{v_0^2}{2g}$	$-\cdot3145\frac{v_0}{2}$
4. θ For Sourc					
5. θ For Sinks					
6. Values of ϕ					
7. Ordt. for ψ					
4. θ For Sourc					
5. θ For Sinks					
6. Values of ϕ					
7. Ordt. for ψ					
4. θ For Sourc					
5. θ For Sinks					
6. Values of ϕ					
7. Ordt. for ψ					
4. θ For Sourc					
5. θ For Sinks					
6. Values of ϕ					
7. Ordt. for ψ					

TABLE II.—STATION 60. [*To face page* 54.

		±0	±10	±20	±30	±40	±50	±60	±70	±80	±90	±100	Ψ values
1. Stations													
2. Abscissæ of Sources..		−60	−50	−40	−30	− 20	− 10	− 0	− 10	− 20	− 30	− 40	
Stations Sinks ..		−60	−70	80	−90	−100	−110	−120	−130	−140	−150	−160	
3. Sink and Source Strength		0	1·9	·6	5·1	6·4	7·5	8·4	9·1	9·6	9·9	10·0	
4. θ For Sources	0′ out	1·571	1·571	1·571	1·571	1·571	1·571	0	−1·571	−1·571	−1·571	−1·571	1186·6
5. θ For Sinks		1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	1·571	
6. Values of ϕ		0	0	0	0	0	0	−1·571	−3·142	−3·142	−3·142	−3·142	
7. Ordt. for ψ value ..		0	0	0	0	0	0	−13·20	−28·60	−30·20	−31·18	−3·142	
4. θ For Sources	10′ out	1·351	1·334	1·296	1·231	1·105	·810	0	− ·810	−1·105	−1·231	−1·296	1034·0
5. θ For Sinks		1·351	1·375	1·388	1·395	1·400	1·404	1·407	1·409	1·411	1·412	1·413	
6. Values of ϕ		0	−·041	−·092	−·164	− ·295	− ·586	−1·407	−2·219	−2·516	−2·643	−2·709	
7. Ordt. for ψ value ..		0	−·075	−·331	−·836	−1·887	−4·397	−11·81	−20·19	−24·14	−26·20	−27·09	
4. θ For Sources	20′ out	1·152	1·109	1·043	·938	·765	·469	0	− ·469	− ·765	− ·938	−1·043	892·4
5. θ For Sinks		1·152	1·182	1·206	1·217	1·228	1·236	1·241	1·245	1·248	1·251	1·253	
6. Values of ϕ		0	− ·073	− ·163	− ·279	− ·463	− ·767	−1·241	−1·714	−2·013	−2·189	−2·296	
7. Ordt. for ψ value ..		0	− ·139	− ·586	− ·422	− ·961	−5·750	−10·42	−15·59	−19·30	−21·66	−22·96	
4. θ For Sources	30′ out	·963	·909	·832	·718	·551	·314	0	− ·314	− ·551	− ·718	− ·832	765·8
5. θ For Sinks		·963	1·001	1·031	1·047	1·061	1·072	1·080	1·086	1·091	1·094	1·095	
6. Values of ϕ		0	− ·092	− ·199	− ·329	− ·510	− ·758	−1·080	−1·400	−1·641	−1·812	−1·927	
7. Ordt. for ψ value ..		0	− ·175	− ·716	−1·678	−3·262	−5·690	−9·060	−12·72	−15·76	−17·95	−19·27	
4. θ For Sources	40′ out	·787	·731	·654	·548	·405	·223	0	− ·223	− ·405	− ·548	− ·654	654·8
5. θ For Sinks		·787	·829	·863	·880	·896	·908	·918	·924	·929	·933	·936	
6. Values of ϕ		0	− ·098	− ·209	− ·332	− ·491	− ·685	− ·918	−1·147	−1·334	−1·481	−1·590	
7. Ordt. for ψ value ..		0	− ·186	− ·751	−1·695	−3·410	−5·140	−7·710	−10·41	−12·80	−14·69	−15·90	
4. θ For Sources	50′ out	·631	·578	·508	·418	·302	·163	0	− ·163	− ·302	− ·418	− ·508	529·6
5. θ For Sinks		·631	·671	·704	·722	·738	·750	·760	·767	·772	·776	·779	
6. Values of ϕ		0	− ·093	− ·196	− ·304	− ·436	·587	− ·760	− ·930	−1·074	−1·194	−1·287	
7. Ordt. for ψ value ..		0	− ·177	− ·705	−1·550	−2·786	−4·405	−6·380	−8·460	−10·31	−11·82	−12·87	
4. θ For Sources	100′ out	0	0	0	0	0	0	0	0	0	0	0	0
5. θ For Sinks		0	0	0	0	0	0	0	0	0	0	0	
6. Values of ϕ		0	0	0	0	0	0	0	0	0	0	0	
7. Ordt. for ψ value ..		0	0	0	0	0	0	0	0	0	0	0	

TABLE III.—Stream Line 20 Units Out at Infinity.

Station	0	30	50	60	70	80	90	100	110	130	160	200
$\frac{ds}{dy}$	2·205	1·970	1·565	1·295	·920	·475	·085	−·290	−·385	−·210	−·105	−·035
$k\frac{ds}{dy} = ·10614\,v_0$ ..	$·2340\,v_0$	$·2091\,v_0$	$·1661\,v_0$	$·1374\,v_0$	$·09764\,v_0$	$·0504\,v_0$	$·0090\,v_0$	$·0308\,v_0$	$·04085\,v_0$	$·0223\,v_0$	$·0111\,v_0$	$·0037\,v_0$
v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0
v_x	$1·2340\,v_0$	$1·2091\,v_0$	$1·1661\,v_0$	$1·1374\,v_0$	$1·0976\,v_0$	$1·0504\,v_0$	$1·0090\,v_0$	$·9692\,v_0$	$·9592\,v_0$	$·9777\,v_0$	$·9889\,v_0$	$·9963\,v_0$
Sec θ	1·0000	1·0022	1·0065	1·0083	1·0094	1·0099	1·0097	1·0072	1·0038	1·0006	1·0000	1·0000
Total $v = v_x$ Sec θ ..	$1·2340\,v_0$	$1·2118\,v_0$	$1·1738\,v_0$	$1·1466\,v_0$	$1·1079\,v_0$	$1·0606\,v_0$	$1·0187\,v_0$	$·9764\,v_0$	$·9628\,v_0$	$·9782\,v_0$	$·9889\,v_0$	$·9963\,v_0$
v^2	$1·5222{v_0}^2$	$1·4681{v_0}^2$	$1·3770{v_0}^2$	$1·3145{v_0}^2$	$1·2271{v_0}^2$	$1·1246{v_0}^2$	$1·0374{v_0}^2$	$·9533{v_0}^2$	$·9270{v_0}^2$	$·9569{v_0}^2$	$·9782{v_0}^2$	$·9926{v_0}^2$
${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$	${v_0}^2$
${v_0}^2 - v^2$	$-·5222{v_0}^2$	$-·4681{v_0}^2$	$-·3770{v_0}^2$	$-·3145{v_0}^2$	$-·2271{v_0}^2$	$-·1246{v_0}^2$	$-·0374{v_0}^2$	$·0467{v_0}^2$	$·0730{v_0}^2$	$·0431{v_0}^2$	$·0218{v_0}^2$	$·0074{v_0}^2$
Pressure from normal	$-·5222\frac{{v_0}^2}{2g}$	$-·4681\frac{{v_0}^2}{2g}$	$-·3770\frac{{v_0}^2}{2g}$	$-·3145\frac{{v_0}^2}{2g}$	$-·2271\frac{{v_0}^2}{2g}$	$-·1246\frac{{v_0}^2}{2g}$	$-·0374\frac{{v_0}^2}{2g}$	$·0467\frac{{v_0}^2}{2g}$	$·0730\frac{{v_0}^2}{2g}$	$·0431\frac{{v_0}^2}{2g}$	$·0218\frac{{v_0}^2}{2g}$	$·0074\frac{{v_0}^2}{2g}$

EAM LINE 20 UNITS OUT AT INFINITY.

70	80	90	100	110	130	160	200
·920	·475	·085	−·290	−·385	−·210	−·105	−·035
·09764 v_0	·0504 v_0	·0090 v_0	·0308 v_0	·04085 v_0	·0223 v_0	·0111 v_0	·0037 v_0
r_0	v_0	v_0	v_0	v_0	v_0	v_0	v_0
1·0976 v_0	1·0504 r_0	1·0090 v_0	·9692 v_0	·9592 v_0	·9777 v_0	·9889 v_0	·9963 v_0
1·0094	1·0099	1·0097	1·0072	1·0038	1·0006	1·0000	1·0000
1·1079 v_0	1·0606 v_0	1·0187 v_0	·9764 v_0	·9628 v_0	·9782 v_0	·9889 v_0	·9963 v_0
1·2271 v_0^2	1·1246 v_0^2	1·0374 v_0^2	·9533 v_0^2	·9270 v_0^2	·9569 v_0^2	·9782 v_0^2	·9926 v_0^2
v_0^2	v_0^2	v_0^2	v_0^2	v_0^2	v_0^2	v_0^2	v_0^2
−·2271 v_0^2	−·1246 v_0^2	−·0374 v_0^2	·0467 v_0^2	·0730 v_0^2	·0431 v_0^2	·0218 v_0^2	·0074 v_0^2
−·2271 $\frac{v_0^2}{2g}$	−·1246 $\frac{v_0^2}{2g}$	−·0374 $\frac{v_0^2}{2g}$	·0467 $\frac{v_0^2}{2g}$	·0730 $\frac{v_0^2}{2g}$	·0431 $\frac{v_0^2}{2g}$	·0218 $\frac{v_0^2}{2g}$	·0074 $\frac{v_0^2}{2g}$

DISCUSSION.

The Secretary read the following communication from—

Sir Philip Watts, K.C.B., LL.D., D.Sc., F.R.S. (Vice-President): I think Mr. Baker and Mr. Kent are to be congratulated on this paper for the general excellence of the research work carried out by them on this very important investigation. It must be a matter of much gratification also to the Members of this Institution that the William Froude Tank has so early brought forth fruit of the highest order of importance. This is the second research paper which emanates from the Tank Superintendent this year, and I understand there are others in preparation; I trust we shall have many such papers.

I gather that two main facts are elucidated in the paper:—

(1) A new law indicating the manner in which the resistance due to transverse wave-making varies with length, speed, and fineness of a ship;

(2) An application of this law to enable approximately the most suitable lengths of parallel middle body to be chosen for a given form and speed.

The manner in which the investigation has been conducted is worthy of note. The authors first conducted an elaborate investigation on the streamline theory to obtain approximately the distribution of pressure around forms of various fineness. Their conclusion, which seems to be justified by the experiments made, is that the relative positions of humps and hollows on the pressure curves vary directly as length and prismatic coefficient, and the argument is therefore that the wave-making effects vary in the same manner. The paper refers to a term which is new to me in this connection, and I should like to ask what is meant by "wave-making speeds" which occurs in § 7 and elsewhere. If this law be true, it obviously applies to the humps and hollows which occur on all resistance curves, and therefore it would be expected that these should be governed by the same law.

It is to be expected that, since the wave-making varies as $P \times L$ (§ 9), the velocities at which humps occur would vary as $\sqrt{P.L}$ in accordance with the law of comparison; but that this connection should be so simple, as explained in the paper, viz., that

$$\frac{V}{\sqrt{P.L}} \text{ should be proportional to } 1, \quad \frac{1}{\sqrt{2}}, \quad \frac{1}{\sqrt{3}}, \quad \&c,$$

seems to be so curious that I would like to ask whether the authors can advance any physical explanation of what seems to be some perfectly natural law.

In referring to particular humps found with destroyers (§ 10), a reference is made to the change in the nature of the bow waves at the topmost hump where $\textcircled{P} = 1{\cdot}5$. It has been suggested that at the high speeds at which such vessels are driven the wave pattern departs considerably from that usually accepted, and I would like to ask whether any such change has been observed, and whether the angle of the diverging waves opened out.

The last section, § 14, is one of great importance in merchant ships, and here in the concluding paragraph it is suggested that the most economical ships are those in which the length of parallel middle body is given by

$$\textcircled{P} = {\cdot}746 \frac{V}{\sqrt{P.L}} = \frac{2}{\sqrt{3}}, \quad \frac{2}{\sqrt{7}}, \quad \frac{2}{\sqrt{11}},$$

while the worst results are given by

$$\textcircled{P} = \frac{2}{\sqrt{5}}, \quad \frac{2}{\sqrt{9}}, \quad \&c.$$

Here again it seems remarkable that the (P) values should vary so that the denominator is an odd number, and I should like to know whether there is any physical explanation for this, and whether the authors have applied this result to any other cases than that quoted, and with what results ?

Mr. A. F. Yarrow (Vice-President) : My Lords and Gentlemen, I think the Institution is under a great obligation to Mr. Baker for bringing forward this paper. On the face of it, it shows a great deal of valuable and steady work, and when we get a paper of this kind we must take care to make use of it. In the experimental work that we do we very often find, and I daresay other people find the same thing, that we carry out a series of experiments, we study them, and then we double them up and put them away and do not make the use of them that we ought. In designing a vessel we often adopt certain lines and certain models. Then someone comes along and proposes that a bit be put on here and a bit taken off there. We are guided mainly by our eyes, but our eyes require to be educated, and it is by papers such as Mr. Baker and Mr. Kent have presented to us to-day that we can educate our eyes on proper lines. What Mr. Baker tells us about parallel bodies is very cheering, because every shipbuilder will be delighted to hear that he can have a number of frames bent off at the same time and off the same mould.

In connection with the importance of testing the resistance of vessels such as Mr. Baker has described, I am informed that Mr. Baker has tested a very large number of models for private firms. He is not, of course, at liberty to divulge the names of these firms, but I think he has suggested improvements in the lines which have resulted in reduction of the resistance ; it would be very valuable if he could give us some idea what he considers to be the average reduction of resistance which he has been able to suggest by alterations in the lines, assuming the length is unaltered, the draught the same, and the displacement the same. If Mr. Baker can give us some idea of what is the average reduction in the resistance that he has been able to suggest, I am sure it would be very interesting information to the Institution.

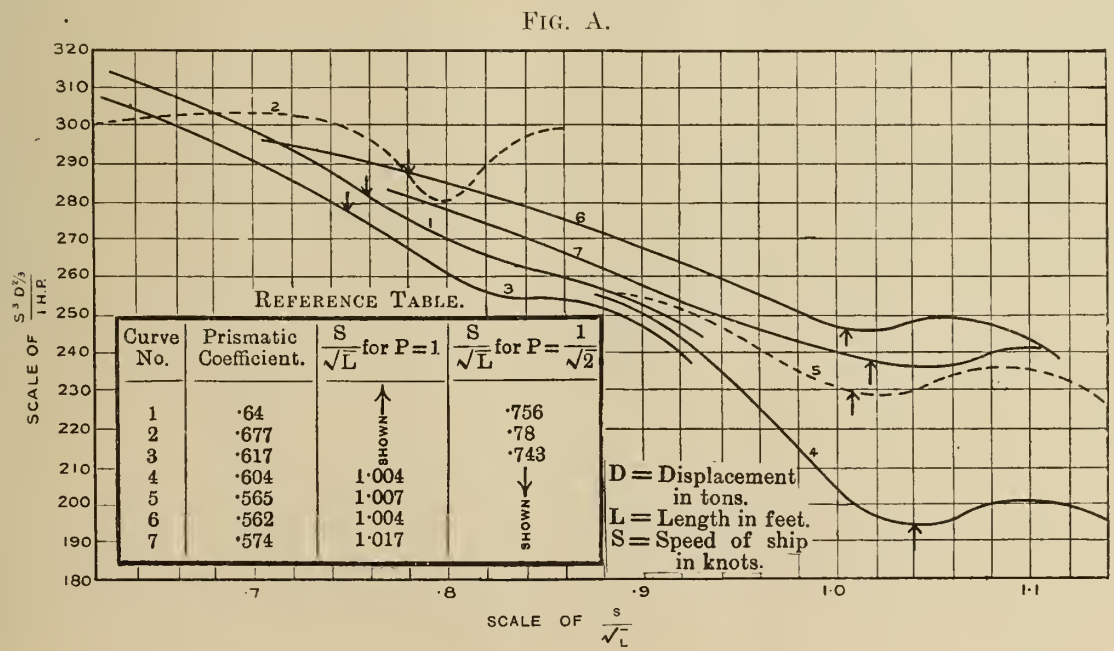
Mr. Max Wurl (Member) : My Lord and Gentlemen, the present paper is certainly a valuable contribution to our knowledge of the laws of ship resistance. We must, however, ask ourselves how far we can apply in practice these formulæ derived from tank experiments ? I have been trying for many years to obtain trial-trip records showing the resistance humps which were known, of course, long ago. After many futile attempts to eliminate the common errors in trial-trip observations, I have succeeded in some cases where it was possible to make a sufficient number of runs on the measured mile under conditions sufficiently favourable for accurate measurement. The curves of Admiralty coefficients (Fig. A) thus obtained show clearly a hump at a speed of about 1·02 to 1·04 of $\sqrt{L}$, and this hump corresponds fairly well with that for (P) = 1 in the paper. The other hump, which corresponds with a greater number of waves, and therefore appears at a lower speed, is sufficiently noticeable with ships of fuller form; but, unfortunately, this hump does not appear at a speed which corresponds with the formula given in the paper ; in fact, if we applied the formula given at the end of the paper for most economical speeds, we would in some cases come just on to the spot of maximum resistance. Possibly the authors of the paper will be able to give some explanation of this discrepancy ; it may be due to the fact that these ships are self-propelled, while the formula is derived from towing experiments on models without propellers. Generally speaking, I think that the interference between bow waves and stern waves has a greater influence in producing maximum resistance humps than the other causes mentioned in the paper, and that consequently the distance actually measured between these two wave crests is the best guide for ascertaining the critical speeds. It seems advisable to continue studying this problem further before applying the above formula too generally in practice.

Mr. A. M. Robb (Associate-Member): My Lord and Gentlemen, I trust you will grant me your indulgence on my first appearance at the meetings of this Institution. In order to get a comparison with Mr. Taylor's experimental results on humps and hollows I have worked out the speed-length ratios at which humps are found on the resistance curves for the three mercantile forms given in Fig. 2 (Plate III.). For No. 1, prismatic coefficient ·827, the hump occurs at a speed-length ratio of ·7. For No. 2, prismatic coefficient ·761, the hump occurs at speed-length ratios of ·7 and ·83. Mr. Taylor states that for quite full models the hump for a speed-length ratio of ·8 is often important, and that for a speed-length ratio of ·67 to ·7 it is often detected. Mr. Taylor unfortunately does not state explicitly what he means by a full model, but Messrs. Baker and Kent's figures for these two models seem to be in close agreement with Mr. Taylor's. I have here assumed that by the expression at the foot of § 9 "the change over from the abnormal hump to the succeeding hollow," Messrs. Baker and Kent indicate a maximum value on the resistance curve. This interpretation seems to me to be borne out by reference to Fig. 2 at the value $\textcircled{P} = 1$, and also by the agreement with Mr. Taylor's results. For model No. 3, prismatic coefficient ·621, the humps occur at speed-length ratios of ·75 and 1·06. Mr. Taylor's results show that for fine models the first important hump occurs at a speed-length ratio of 1·4 to 1·5, there being a slight hump at a speed-length ratio of 1. The latter figure corresponds very closely to Messrs. Baker and Kent's figure of 1·06. In passing, it seems to me that the expression in § 9 is very deceptive. At first glance it would seem to indicate that there is an advantage in having a high prismatic coefficient. There is also a point in connection with § 14 on which I should like further information. In § 9 we find that a hump occurs at a speed given by $V = 1{\cdot}34 \sqrt{P \cdot L}$. In § 14 we find that a mean resistance line is given by

$$\textcircled{P} = {\cdot}746 \frac{V}{\sqrt{P \cdot L}} = 1.$$

Now $V = 1{\cdot}34 \sqrt{P \cdot L}$ and $\textcircled{P} = 1$ are different forms of the same expression. What I should like

Fig. A.



Reference Table.

Curve No.	Prismatic Coefficient.	$\frac{S}{\sqrt{L}}$ for P=1	$\frac{S}{\sqrt{L}}$ for $P=\frac{1}{\sqrt{2}}$
1	·64	↑ SHOWN	·756
2	·677		·78
3	·617		·743
4	·604	1·004	↓ SHOWN
5	·565	1·007	
6	·562	1·004	
7	·574	1·017	

the authors to explain is why the expression in § 9 gives us the speed at which there is an important hump and the same expression, or another form of it, seems to give us a mean resistance line.

Mr. W. H. Riddlesworth, M.Sc. : My Lord and Gentlemen, on examining this paper of Messrs. Baker and Kent, I and many others have again turned back to William Froude's paper of 1877, and on examining the speeds, or rather the lengths at which the hollows in the resistance curves given there occur, and applying the author's formula to the corresponding speeds, I find quite considerable differences. I have a few of them worked out here, and with your permission I will try and quote them. Take the case of the series of ships that were tested at a speed of 14·43 knots. The diagram is reproduced in the present paper in Fig. 8 (Plate IV.). The first hollow occurs at a length of 168½ ft. from the bow, or 8½ ft. from end of scale. The authors' formula gives something less than 160 ft., how much I cannot say, because it cannot be applied in this case. The next hollow occurs at a length of 277 ft. The authors' formula gives 276. The next one is 385 ft., and the formula gives 391 ft. There is a curious difference between the two sets of results, a positive difference at the first hollow, a practical coincidence at the second, and a negative difference in the last one. If you take the next lower speed, at which you can get three hollows, the first hollow on the Froude curve is 164 ft., and the authors' formula shows something less than 160 ft.

Mr. Baker : I think Mr. Riddlesworth is not using the formula properly. The formula given in the paper is for a given boat, giving the hump in the resistance curve for that boat. The curve for 14·43 knots in Mr. Froude's experiments is the curve of resistance for a number of boats which all differ in their length. If you examine the last diagram of the paper it will show you that these hollows at 14·43 knots come, as you would expect them, at a quarter the space from the one (P) curve to the next (P) curve. The diagram represents a multiplicity of boats of varying length with humps which have a natural spacing with regard to the humps for any individual form run at increasing speeds, *i.e.* a cross curve of Mr. Froude's parallel body series. The formula is not intended to give the humps on Fig. 8 of the paper, as suggested by Mr. Riddlesworth.

Mr. Riddlesworth : That indicates that my basis of examination has been such as was not contemplated by the authors of the paper, and therefore does not apply.

Mr. G. S. Baker (Member) : My Lord and Gentlemen, I appreciate very much Sir Philip Watts' kind remarks on the paper. With regard to the practical use of the law, of course, some common-sense is required in its application. The designer has to learn—and can easily learn by model experiments —what is the highest (P) value to which he can go with any design without the magnitude of the hump in the resistance curve becoming excessive. He also has to take care that the entrance wave is not going to give trouble at the ship's service speed. Sir Philip asked what is meant by "wave-making speeds." The term "wave-making length" would, perhaps, be better, as the subject referred to is closely allied to the "statical wave" used by Mr. Froude in his Greenock paper. The physical explanation of the resistance humps has been given by Mr. Froude. When the crest of any transverse waves formed by the bow comes at the same fore and aft position as the crest of the wave which the stern is tending to form, there will be abnormal resistance. When the second crest of the bow system comes over the first crest of the stern system, the velocity is given by (P) = 1·5. If the first crest of the stern system comes at mid-height of the forward slope of the third crest of the bow system, increase in speed will improve results, and decrease in speed will make them worse. This speed is given by (P) = 1·0. The other (P) values $\frac{1}{\sqrt{2}}$, $\frac{1}{\sqrt{3}}$, are intended to be measures of the speed when the same conditions hold with an increased

number of waves in the bow system. The actual (P) values are empirical, differing slightly on the low side from the theoretical values, but they are simple to remember and quite accurate. We have applied the formula to a large number of cases, from submarines to shallow draught steamers, and have never found it to fail when the hump has been really noticeable.

With regard to Mr. Yarrow's question, average figures are liable to be misunderstood, as so much depends upon the goodness or badness of the original design. One of the best examples of the use of the tank was that of a merchant vessel sent to us to test. By the use of the formula, and our general knowledge of ship resistance, we were able to increase the parallel body from 8 per cent. to nearly 30 per cent., slightly modify the general form, leave the over-all dimensions unaltered, and decrease by 15 per cent. the horse-power required to drive the ship. So far, we have been able to suggest improvements in the lines of every model sent to us for testing, and these improvements have resulted in a reduction of power of at least 3 per cent. This means that in no case has a model been sent to us for testing in which the shipowner has not recovered the cost of the test by saving on the coal bill of the ship in the first six months of its running.

In reply to Mr. Robb's remarks, we have not compared our formula with any formula given by Mr. Taylor, but as we have found it to apply to all the model results published by Taylor, one would expect to find the agreement which Mr. Robb has found.

* Mr. Wurl's diagram of ship-trial results is of considerable interest, as in some of the curves which it shows, the resistance hump is well marked. The speeds for $(P) = 1$ or $\frac{1}{\sqrt{2}}$ are marked upon these curves, and it will be seen that they come at a speed about 1 to 3 per cent. lower than those given by the trials. This discrepancy is small, and, neglecting the possibility of inaccuracy in the curves, there are two possible explanations of the difference :—

(1) As suggested to us by Mr. Wurl, it may be due to the action of the propellers.

(2) It may be due to variation of screw efficiency.

It is quite possible that the propellers may cause a very slight increase in the length of the ship's statical wave, and thus raise the speed at which the various humps and hollows occur. It is equally possible, and more probable, that the screw efficiency is falling off with higher speeds, which would tend to show the I.H.P. hump at a higher speed than that at which it would occur on the towed ship.

Dr. Gümbel (whose remarks are given below) is pleading for unification in the nomenclature of ship resistance. There can be no doubt that it would be an advantage if all the data *re* ship resistance were published in terms of abstract constants. The Froude constants are intended to be abstract figures using the English system of units, and some of them are the same in any system, metric or English. From the purely scientific point of view the metric system is much simpler, but from the practical point of view it has no advantages, and it is not difficult in any particular case to turn results from one system into another.

The Chairman (the Right Hon. Lord Inverclyde, Associate Member of Council) : I am sure, in your name, I may thank Mr. Baker and Mr. Kent very heartily for the very satisfactory and interesting paper that they have brought before us.

Written Contribution.

Dr. Gümbel (Member) : I regret not having been able to attend personally the reading of Mr. Baker's very interesting paper, as I was at that time engaged with my own paper on the Cavitation Problem in

* Added after the Meetings.

the University Lecture Room. I am convinced that an ample discussion will bring to light the importance of Mr. Baker's contribution to the problem of ship resistance.

There is one point in the paper which appears of considerable interest to me, and to which I should like to draw your attention. In his researches, Mr. Baker makes use of the Froude constant

$$\text{Ⓒ} = C_1 \frac{EHP}{\Delta V^3}$$

and of the value

$$\text{Ⓟ} = C_2 \frac{V}{\sqrt{PL}}$$

He proves that these formulæ are very useful for reproducing in a uniform way the results of model experiments. But—and this is the point—these terms are not convenient when the metric system is used instead of the English system, and when the fluid in question is of a different specific gravity from that of the water in the tank. Fortunately, small changes in the above terms can be introduced into both the metric and English systems, and for any fluid, say, for salt or fresh water, or for air, by altering the terms

$$\text{Ⓒ} = C_1 \frac{EHP}{\Delta^{\frac{2}{3}} V^3} \quad \text{to} \quad C_3 \frac{Rg}{\Delta^{\frac{2}{3}} w^{\frac{1}{3}} V^2}$$

and

$$\text{Ⓟ} = C_2 \frac{V}{\sqrt{PL}} \quad \text{to} \quad C_4 \frac{V}{\sqrt{PLg}}$$

R being the resistance of the ship or model in the unit of force, say, kilogrammes, or pounds or tons; Δ being the displacement in the unit of force, say, kilogrammes, or pounds or tons; w being the weight per unit of volume, say, kilogrammes per cubic metre, or pounds or tons per cubic feet; L the length of model or ship in the unit of length, say, metre or foot; g the acceleration of the earth in suitable units; V the speed in unit of length per second, say, metres or feet per second. Then the two terms will be abstract figures having no dimensions at all, and consequently they will have the same value in both the metric and English systems, and for fluids of any specific gravity.

In my opinion, it is to be deeply regretted that in the technical world two different systems—the metric and the English—should still be in use, thus preventing the immediate application of results obtained by experimenters in different countries. This appears to me reason enough to call special attention to any method that will remedy this evil, and establish a common international basis for researches of the importance of those described in Mr. Baker's paper.

EXPERIMENTS ON "SUCTION" OR INTERACTION BETWEEN PASSING VESSELS.

By Professor A. H. GIBSON, D.Sc., and J. HANNAY THOMPSON, Esq., M.Sc.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 25, 1913; the Right Hon. Lord INVERCLYDE, Associate Member of Council, in the Chair.]

1.—THE experiments to be described in this paper have been carried out with a view of obtaining some information as to the magnitude and range of action of the forces involved in the case of "suction" or interaction between passing vessels.

Up to the present time, in such experiments as have been performed, models of comparatively small size have been used, and while these have given extremely valuable results, some doubt has been expressed as to the extent to which their results are susceptible of extension to the case of vessels of large size.

In the present series of experiments two screw propelled vessels were used. One of these, the steam yacht *Princess Louise*, is 88·5 ft. in length, 13 ft. beam, 5·66 ft. mean draught, displacing approximately 96 tons. The second is a motor-driven launch, 29·33 ft. long, 6·75 ft. beam, 1·37 ft. mean draught, displacing approximately 2·6 tons. Each is driven by a single screw.

The experiments were divided into two distinct sets. In the first the vessels were manœuvred until on sensibly parallel courses, heading for the same distant object; their lateral distances apart and speed being varied in different experiments.

The courses having been satisfactorily fixed, with the helm of the motor boat amidships, this helm was lashed, the helm of the *Princess Louise* being afterwards manipulated so as to keep her on her original course.

Two plane tables with alidades were mounted on the deck of the *Princess Louise*, distant 67 ft. 8 in. centre to centre, and the relative position of the motor boat was fixed at intervals of fifteen seconds during each run by means of simultaneous sights taken from these. Both vessels were calibrated on the measured mile before the experiments, and speed-revolution curves were obtained from which, by counting the revolutions, the speed of each vessel could be ascertained or regulated.

These data enabled the relative positions and paths of the two vessels to be plotted with a close degree of accuracy, and the diagrams illustrating the paper have been obtained in this way.

With a view of measuring the forces and moments involved, a series of eight pressure boxes were fixed to the hull of the motor boat at about 12 in. below the water-line. These are circular; have a diameter of $1\frac{1}{2}$ in., and a maximum thickness of $\frac{3}{8}$ in., and communicate with the sea through four $\frac{1}{16}$ in. holes on a 1 in. circle. They are in pairs at similar positions on the port and starboard sides, and distant 4, 10, 16, and 22 ft. from the bows. Each corresponding pair was attached by means of rubber tube connections to the branches of one of a series of inverted "U" tubes carried on deck, and by exhaustion of the air from the top of the tubes a column of fluid from each pressure box was brought on to a graduated scale, from which the differences of pressures at corresponding points were determined. Each pair was calibrated at different speeds so as to give the zero reading with the boat travelling on a straight course and remote from any disturbing influence.

Readings of the gauges were taken each fifteen seconds in a number of the experiments, and from these the turning moments and lateral forces acting on the hull of the boat have been computed.

The second series of experiments was carried out with a view of measuring the helm angle required to maintain the course of the smaller vessel, when in the vicinity of the larger one.

In these experiments the relative positions of the two boats were obtained as before, the helm being adjusted as required to keep the head of the vessel on its original course. The helm angle was measured by means of a pointer fixed to the tiller, and working over a graduated sector, and was observed at intervals of 15 seconds.

The rudder originally fitted to the motor boat was proportionately about three times as large as is fitted to the average sea-going steamship. This was replaced for the purpose of the experiments by a rudder of 144 sq. in. area, for which one of 75 sq. in. area was afterwards substituted, the latter representing approximately to scale the rudder fitted to the average large steamship. The results of control experiments using each rudder in turn are given at a later stage of the paper.

Owing to the risks involved in the collisions, the speed of the vessels was restricted to a maximum of about 5·75 knots, which, in the case of the *Princess Louise*, corresponds to 18 knots in a vessel of the dimensions of the *Olympic* (882 ft. long). The minimum speed was about 3·5 knots.

Owing to the local conditions it was found impracticable to carry out the experiments in shallow water of even approximately constant depth, and, except in

two experiments (Nos. 41 and 42, Plate VII.), where the water was about 12 ft. deep, the depth actually ranged from 20 to 30 ft. Since this is from 12 to 20 times the mean draught of the smaller vessel, these are essentially deep water experiments, and, as is indicated both by theory and experiment, the forces involved are in general less than would be experienced in shallow water.

2.—General Theory of Interaction.

When a ship-shaped body is towed through still water, both theory and experiment show that a general circulatory motion is set up in the surrounding fluid. Those particles ahead of the bows are first affected, being forced forwards and outwards into a fan-shaped region extending ahead from the bows, in which the pressure and elevation are greater than normal. From this region a general flow takes place, backwards and inwards to fill the space vacated by the stern. As the velocities of flow increase the pressure diminishes, to become a minimum abeam of the body, while as, due to convergence at the stern, the velocities diminish, the pressure again increases. The body thus carries along with it a region of depression abeam and fan-shaped regions of excess pressure and super-elevation respectively ahead of its bows and in the rear of its stern.

The pressures and velocities of flow diminish rapidly as the distance from the vessel increases, the pressures varying as the square of the corresponding velocities. They are greatest in a narrow channel where the area available for backward flow is restricted, while theory indicates that they are of necessity greater in shallow water in which flow takes place in layers approximately parallel to the surface, than in deep water where inflow and outflow may take place freely in vertical as well as in horizontal directions.

When a vessel is propelled by its own screw the distribution of pressure is somewhat modified. The screw propeller, drawing a large volume of water from regions ahead and abreast of itself and throwing this astern, tends to increase the velocities of flow, to reduce pressures ahead, and to increase those astern of itself.

When two ship-shaped bodies are moving in close proximity, each affects the lines of flow and the distribution of pressure around the other. Those pressures between the hulls suffer the greatest modification, and it is this modification in the distribution of pressure over the two sides of each of the vessels which gives rise to interaction between the two.

If one vessel is overtaking a second one on a parallel path, the effect on the overtaking boat of the system of currents accompanying the leader is broadly as follows:—

As the faster vessel draws up astern it first comes within the influence of the

outflowing currents astern of the leader, and as the strength of these is greater at its bows, it experiences a tendency to sheer off from the leader (Fig. 1*a*).

Creeping further ahead, its bows come within the influence of the inflowing currents while its stern is still being repelled, and it consequently tends to sheer in (Fig. 1*b*). This tendency generally becomes a maximum when the bow of the follower is a little abaft the beam of the leader, the exact relative position depending on the

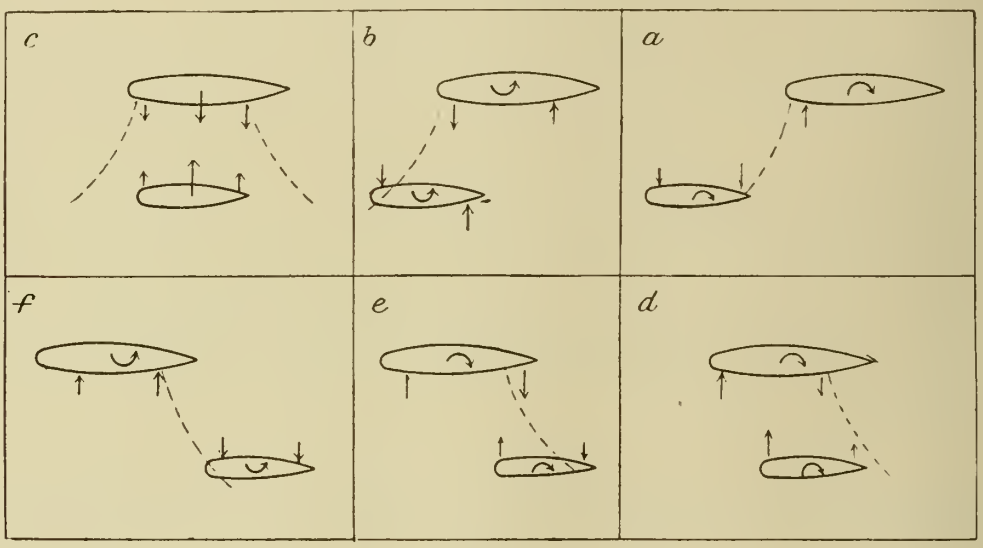


FIG. 1.

relative sizes and distance apart of the two vessels. If the follower passes this point in safety the tendency to inward sheer gradually diminishes until, when a little abaft the beam of the leader, it disappears and is replaced by a strong tendency to bodily inward drift (Fig. 1*c*).

A further advance reduces the inward force on the bows and increases that on the stern, with a consequent tendency to outward sheer (Fig. 1*d*), which is increased when, as in Fig. 1*e*, its bows come within the influence of the outflowing currents from the bows of the leader. Finally, as the follower draws ahead and becomes the leader, both bow and stern come within the influence of these currents, and, those at the stern being greatest, the stern is repelled more strongly than the bows, and the vessel tends to sheer inwards (Fig. 1*f*).

The influence of the currents produced by the follower will, of course, be felt by the leader, and will tend to produce erratic movement on its part. The relative magnitudes of the various forces, and the exact point at which they change sign, varies with the relative sizes of the vessels and with their distance apart, but their general

nature as regards direction and effect on both vessels with ships of the relative sizes and at the relative distances shown is indicated in the diagrams of Fig. 1. The arrows applied to the bow and stern of the vessels in these diagrams show the directions of the resultant forces at each point, and the curved arrows on each vessel indicate the direction of sheer under these forces. The curved dotted lines in the diagrams mark the boundaries between the regions of positive and negative pressure accompanying the leading vessel.

If, instead of being parallel, the overtaking vessel is inclined to the leader in the position (Fig. 1 *b*), the inflowing currents act more directly on its bows, and, at a given distance, their effect will be consequently increased.

Modifying Effect of Bow and Stern Waves.—Any ship-shaped body in motion is accompanied by a train of waves diverging from the bow and stern, and often, in addition, from some one or more points amidships, as indicated in Fig. 2, in which the full lines represent the crests and the dotted lines the troughs of the waves. The exact type of wave formation and the relative positions of crests and hollows with any given ship depend on its speed, and the modifying effect on a second vessel of the distribution of pressure set up by these, depends largely on the relative lengths and positions of the two vessels. Under the circumstances indicated in Fig. 2, however, the effect will be broadly as follows :—

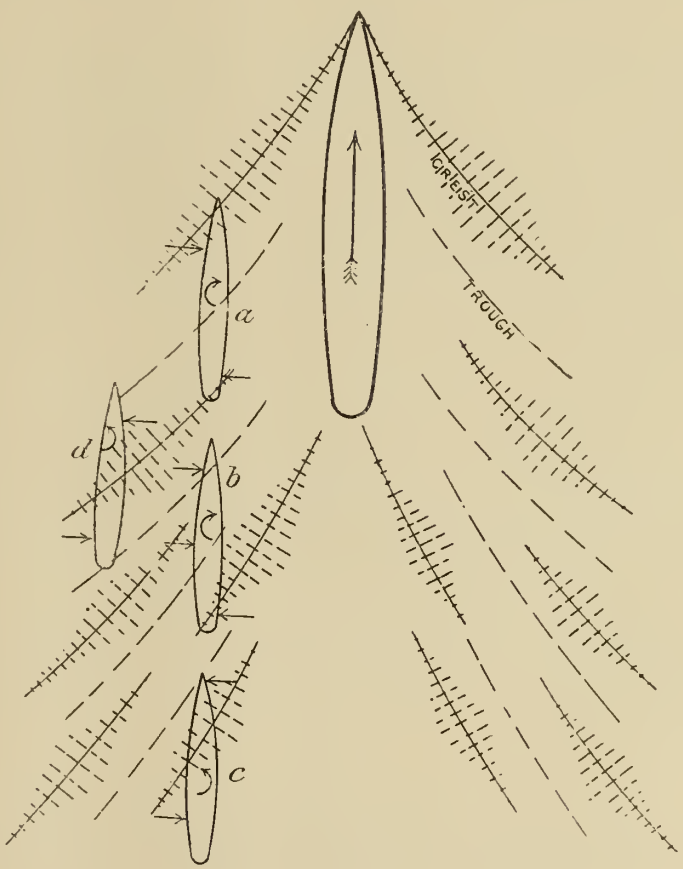


FIG. 2.

If the follower be at *a* the height of water at a given point on its port side near the bows is greater than at the corresponding point on the starboard side, with a consequent resultant pressure to starboard. Near the stern, however, the resultant pressure at corresponding points is to port, and as a result of these forces the vessel

will tend to swerve in towards the leader. The same holds true if the vessel be in the position *b*, while if in position *c* or *d* the reverse is the case. With a longer vessel in the same position relative to the leader, these actions would be modified and might indeed be entirely reversed, while at different speeds, with a different wave formation, the effects would also be different. In any particular case, however, these waves tend to modify the effect of the circulatory currents, increasing it in some positions and reducing it in others.

The difference due to this cause in the lateral forces over corresponding vertical elements of the hull is, however, not nearly so great as would be produced by the same statical differences of level, since the pressure at a given depth below the crest of a wave is less than at the same depth below the trough. At a given point in water through which a train of waves is progressing the pressure becomes more nearly constant and equal to the pressure at the same point in the water at rest, as the depth of the point increases. With a vessel of draught so great that the vertical orbital motion of the particles at the level of its keel is negligible, the resultant lateral force over any vertical element of the hull would indeed be unaffected by the presence of the wave.

The deeper the water, the greater is the vertical motion at a given depth accompanying a given wave formation, and the greater the resultant effect on a vessel of given draught. From this it appears that the wave effect is greatest in deep water, where the effect of interaction is otherwise least, and *vice-versâ*. That the result may be very appreciable with suitable relative lengths of ship and wave is evident when it is realised that a mean effective elevation of 1 in. on the one side from bow to midships, and a mean elevation of the same amount on the other side from midships to stern of a vessel 360 ft. long drawing 20 ft. of water, involves a sheering moment of the order of 1,500 foot tons.

3.—Experimental Results.

Diagrams 1 to 19 (Plates V. and VI.) show graphically the results of typical experiments in which the helm of the motor boat was lashed amidships.

In the experiments shown in diagrams 1 to 8 the speed of the *Princess Louise* was 5·1 knots; in those of diagrams 9 to 13 it was 3·61 knots; while in those of diagrams 14 to 19 it varied from 3·6 to 5·8 knots.

The experiment shown on diagram 1 shows the repulsion when the smaller boat is ahead of the larger, gradually changing into an attraction as the latter boat draws abreast, and finally producing collision from a lateral distance of 3·4 lengths of the smaller boat.

Experiments 2 to 7 show the same attraction occurring at different relative longitudinal positions of the vessels, with lateral distances ranging from 1·7 to 3·5 lengths of the smaller boat.

Initial repulsion is well marked in No. 8, in which the steam yacht *Princess Louise* is the faster vessel. Attraction, however, becomes apparent at a distance of three lengths when the vessels are approximately abeam, and is followed by collision in one minute.

In the experiments 9 to 13 the speed of the *Princess Louise* was 3·61 knots. Experiment 9 is interesting as showing the vessels in relative positions in which, in spite of their close proximity, the resultant moment is approximately zero.

Experiment 10, in which the *Princess Louise* is somewhat faster than the motor boat, shows the repulsion when the smaller boat is slightly ahead, changing to attraction as the larger boat draws abreast, and terminating in collision from a distance of one and a half lengths. The repulsion is also well marked in experiment 11, in which the motor boat is at first slightly the faster.

In experiments 12 and 13 the initial lateral distance is respectively $1\frac{1}{3}$ and $1\frac{2}{3}$ lengths, but the initial relative longitudinal positions are different in the two cases. In each case collision was produced.

In experiments 14 to 19 (Plate VI.) the relative speeds were much greater than in the preceding cases. In Nos. 14, 15, 16, and 17 the speed of the *Princess Louise* was 3·75 knots, while that of the motor boat was respectively 3·9, 4·5, 5·1, and 5·4 knots.

In each case the experiment was commenced with the smaller boat in approximately the same relative position, viz., about two lengths laterally distant, and about a length astern of the leader, and in each case the smaller boat was attracted.

In No. 14 the speed of the vessel was insufficient to allow of its catching up the *Princess Louise* before being drawn in, and it passed under the stern of the latter. In Nos. 15 and 16 collision took place, while in No. 17 the motor boat, although considerably deflected from its course, ran past the *Princess Louise* without collision.

In experiments 18 and 19 the speed of the motor boat was 3·9 knots, and that of the *Princess Louise* was 5·9 knots. At the beginning of both experiments the smaller vessel was clear of the bows of the *Princess Louise*, and at a lateral distance of 1·5 lengths in No. 18 and 2·0 lengths in No. 19. Under these circumstances it was drawn into collision in No. 18, while in No. 19 the *Princess Louise* ran ahead before the deflection of the course of the motor boat was sufficient to produce collision, the latter boat running under the stern of the *Princess Louise*.

The authors would draw attention to the fact that collision, when produced by attraction from a comparatively large distance, is of a much more direct, and consequently dangerous, nature than when the paths of the vessels are initially very close. In the former case the forces involved are operative for a sufficiently long time to produce a comparatively large angular deflection of the attacking vessel, the lateral component of whose own steaming speed becomes increasingly operative in increasing the velocity of approach. In the latter case the angular deflection is comparatively small, the velocity of approach is largely due to the bodily inward drift, and the vessels come together with a comparatively slight and innocuous broadside bump.

4.—Magnitude of Moments and Forces Involved.

As previously indicated, the differences of the pressures obtaining at four corresponding equidistant points on the two sides of the smaller vessel were measured for different positions of this vessel, and the corresponding lateral forces and pressures have been calculated on the assumption that the pressure varied uniformly between consecutive gauges.

Owing probably to the clogging of one or more of the pressure openings by floating material, some of the pressure observations gave obviously inconsistent results during the course of a run, and only those experiments have been included in which these observations were consistent throughout.

Experiments 20 to 24 (Plate VI.) are typical of these, and the relative positions of the vessels in these experiments are shown along with the magnitudes in foot-pound units of the corresponding moments (M) and lateral forces (F).

Only during the latter portions of these runs were the various forces of sufficient magnitude to admit of reasonably accurate measurement by the method adopted, and only where those forces are greater than 10 lbs., and the moments greater than 20 ft. lbs., have these been given in the diagrams.

Experiments 20 and 21 were carried out at a mean speed of 5 knots, and experiments 22 to 24 at 3·61 knots.

The forces and moments acting on the smaller vessel depend on its lateral and longitudinal positions relative to the larger, and on the speeds of the two vessels. Under given speed conditions the sheering moment appears to attain its maximum value when the bows of the smaller are a little abaft the beam of the larger vessel, the position of maximum sheer apparently receding slightly as the lateral distance increases, while when the bows are in line the force and moment are approximately zero. A further advance of the smaller boat is accompanied by an outward force and moment, with consequent repulsion of this vessel.

Both forces and moments diminish rapidly with an increase in lateral distance. For distances between one-half and three lengths the results show that the moment diminishes very approximately as the square of the lateral distance.

Theoretical considerations indicate that forces and moments should increase with the square of the speed of the vessels. Actually in the experiments the rate of increase was slightly less than this. Since the turning moment exerted by the rudder is proportional to the square of the speed, it follows, as was noted in the helm control experiments, that the vessel is somewhat more easily controlled against suction forces at high than at low speeds.

Calculations show that the controlling moments of the rudder in use during these experiments (area 1 square foot) are approximately as follows (measurements in foot lbs.) :—

Speed (knots).	Helm Angles.				
	5°	10°	15°	20°	25°
5·0	100	180	280	340	400
3·6	52	93	145	175	210

From these figures it appears that the maximum moments measured in the experiments, viz., 240 ft. lbs. at 5 knots, and 160 ft. lbs. at 3·6 knots, are such as would be controlled by helm angles of about 13° and 16°, while when at a lateral distance equal to one length of the smaller boat, and in the position of maximum sheer, the helm angle for control would be approximately 8° at 5 knots and 9° at 3·6 knots. These values are very sensibly confirmed by the results of the helm control experiments.

5.—Helm Control Experiments.

In the first helm control experiments a rudder having an area of 1 sq. ft. was used on the motor boat. While considerably less than the rudder originally fitted to the boat, this has an area proportionately about 100 per cent. greater than is usual in high speed sea-going vessels.

It was anticipated that the effect of this would be to diminish the helm angle necessary for control by about the same percentage as compared with a large vessel under similar conditions, and with a view of testing this point a smaller rudder of 75 sq. in. area was afterwards fitted, and was used in a further series of trials. In each run the helms were manipulated so as to keep both vessels as near as possible on parallel courses, heading for the same distant point.

Typical results of experiments with the large rudder are shown in diagrams 25 to 30 (Plate VI.), and with the smaller rudder in diagrams 31 to 37. In these diagrams the speed of each vessel, the helm angle for each position of the motor boat, and the direction (port or starboard) of the helm, has been indicated.

An examination of the results shows that, as indicated by the gauge observations, the maximum sheer is experienced when the bow of the smaller is a little abaft the beam of the larger vessel.

The angle diminishes rapidly as the lateral distance between the vessels increases, and when these are moving at the same speeds (5 knots), and are at a lateral distance equal to one length of the smaller vessel, has a maximum value of approximately 6° with the larger and 9° with the smaller rudder. At a lateral distance of one-half length this increased to about 10° with the larger and 14° with the smaller rudder.

It should be noted that in these trials all initial tendency to swerve was at once corrected by the helm, and was, in fact, anticipated as far as possible. Where such a sheer has once been initiated, and the vessels are travelling on convergent paths, a much greater helm angle is necessary to restore the attracted vessel to its original course. In experiments in which the helm was kept amidships until the two vessels were within one-half length of each other, the angle necessary to prevent collision with the larger rudder was approximately 17° at 5 knots, and 23° at 3·6 knots. It will thus be seen that twice the amount of helm angle is necessary to prevent collision if the boat has once commenced to sheer in. The experiments indicate that, on the whole, the smaller vessel is under better control at the higher than at the lower speeds.

Diagrams No. 34 and 35 (Plate VI.) are interesting, as showing the comparatively large helm angle required to prevent an outward sheer when the smaller is ahead of the larger vessel, whilst Nos. 26, 30 and 37 show the (smaller) angle required to prevent outward sheer when the larger has drawn well ahead of the smaller vessel.

The effect of a variation in the relative speeds of the vessels is well marked in experiments 33 and 34. In the former, the smaller at 5 knots is passing the larger vessel at 3·6 knots, with the result that when a little abaft the beam of the latter vessel and at a distance of one length, 2° of helm is sufficient for control.

In No. 34, the larger vessel at 5 knots is passing the smaller at 3·6 knots, and when in the same relative position, 10° of helm is necessary for control. When both are moving at the same speed an angle of approximately 7° is necessary in this position. It follows that any attempt of the larger vessel to draw ahead of the smaller by increasing the speed greatly increases the risk of collision.

A somewhat surprising feature of the results is the comparatively small difference between the control exerted by the two rudders when under the influence of interaction. In the open the difference in their effect, as shown by the difference in the two turning circles, was very apparent, and the difference in the two sets of conditions is probably largely due to the fact that when in the vicinity of the larger vessel the

greater turning moment of the larger rudder is partly masked by the more rapid inward drift consequent on the greater lateral pressure on its surface.

There is reason to anticipate that the tendency to sheer, as opposed to bodily drift, would be greater with an attracted vessel of greater relative dimensions than that used in the experiments. This vessel is in a field of force whose intensity varies from point to point, and which consequently gives rise to both a resultant force and a couple. The average magnitude of the force per unit area of the side of the vessel depends solely on the mean intensity of the field extending from stem to stern, and is, therefore, largely independent of the length of the vessel. It follows that the resultant force is increased in the same ratio as the area of the vessel, and, with similar vessels, in the ratio of the squares of corresponding linear dimensions. Since the weight of the vessel is increased in the ratio of the cubes of such dimensions, the resultant force per unit of mass of the vessel increases directly as the inverse of the linear dimensions, and, consequently, the smaller the attracted vessel the more rapid will be its inward drift.

On the other hand, as the size of this vessel is increased the differences between the forces exerted at bow and stern become greater, and the resultant couple is increased in a greater proportion than the resultant force. As a consequence, the smaller the attracted boat, the less relatively is its tendency to turn inwards about its own axis. With a very small boat, collision, when it does occur, is mainly due to the inward drift; the velocity of lateral approach is low, as it is only slightly affected by the steaming speed of this vessel, and the resultant impact is at a comparatively small angle.

This conclusion receives some confirmation from the results of experiments on 20 ft. models of equal size, by Naval Constructor D. W. Taylor, of the U.S. Navy.* In these experiments the models were attached to a carriage and were towed at the same speed. They were maintained in definite relative positions during a run, and the lateral forces required at bow and stern to prevent swerve were then measured by spring dynamometers.

A comparison of these results with those of the Authors' shows that the proportional reduction in the resultant lateral force with the two smaller but equal vessels is much greater than the reduction in the sheering moment.

6.—Effect of an Increased Draught of the Attracted Vessel.

With a view of determining the effect of an increase in the draught of the attracted vessel an additional series of experiments has been carried out during the past few

* Trans. Amer. Soc. Naval Arch., Vol. XVII, page 1.

weeks. For these a deeper keel was fitted to the motor boat, increasing its mean draught from 1·37 ft. to 2·87 ft. without sensibly affecting its displacement.

Diagrams 38 to 42 (Plate VII.) show the results of typical experiments on the vessel with the helm amidships, Nos. 38 to 40 being carried out in deep water, and 41 and 42 in water having a depth of about 12 ft.

A comparison of Nos. 38 and 39 with Nos. 7 and 3 (Plate V.), which are very similar as regards speed and lateral distance, appears to indicate that the effect of the increased draught is slight, and that collision is produced within about the same time in each case. This is, however, to be qualified by the fact that in runs 38 and 39 the vessel was exposed to a breeze on the port bow, equivalent, as was shown afterwards, to a helm of about 3°, so that in effect, in these two experiments, a constant helm of roughly this amount was tending to keep the vessels apart. In experiment 40 (compare No. 4, Plate V.) the smaller vessel was under the lee of the larger, and in this case the attraction is notably greater than with the shallower keel.

In experiments 41 and 42, in which the depth of water was only about one-half that in the other experiments, the general opinion of the observers, confirmed by the plotted results of the experiments, was that the attraction was somewhat greater than in the deeper water, although the difference was not very pronounced.

Diagrams 43 to 46 (Plate VII.) show the results of helm control experiments, in which the same rudder (1 sq. ft. area) was used as in experiments 25 to 30 (Plate VI.).

A comparison of these two sets of experiments shows that the helm required to prevent swerve is very appreciably greater with the deeper keel—about twice as great, in fact, as with the shallow keel under otherwise similar circumstances. At equal speeds (5 knots) at a lateral distance of half a length in the position of maximum shear, a helm angle of about 17° is necessary to prevent sheer as against 10° with the shallow-keeled vessel, while where the helm is kept amidships until within this distance a helm of 35° was sometimes insufficient to prevent collision.

General Conclusions.

In general, the greater the difference between the speeds of the vessels the smaller is the risk of collision, since such a difference reduces the time during which the mutual forces are operative; such an effect being much more marked when the smaller vessel is the faster. If the larger vessel is the faster, particularly if her speed be accelerated while passing the smaller, the attractive forces are increased to an extent which partially, and in some cases entirely, counterbalances the effect of the reduction in

the time during which the vessels are in dangerous proximity. It follows that any attempt of the larger vessel to draw ahead of the smaller by increasing her speed, while in close proximity, greatly increases the risk of collision.

With vessels of the relative size used in these experiments, moving at speeds within 10 per cent. of each other, collision may be produced from a lateral distance as great as $3\frac{1}{2}$ lengths of the smaller vessel, except in so far as prevented by helm action.

The greater the draught of the attracted vessel, for a given displacement and length, the greater the probability of ultimate collision.

The smaller the attracted vessel, within limits, the smaller is the angle of impact under given conditions, while the greater the lateral distance from which collision is produced, the more direct and dangerous is the resultant impact.

On the whole, the results of the trials show that under certain circumstances interaction is a very real danger to navigation, even in deep and open waters.

With ordinary vessels of the relative sizes adopted for the experiments, if the possibility of interaction is realised from the very first, and if all initial swerve is prevented by an early application of the helm, there would appear to be little danger even at lateral distances so small as one-half the length of the smaller vessel, but once such a swerve has been initiated a much greater helm angle is necessary to control the vessel, and, failing immediate control, collision occurs within comparatively few seconds.

Note added after the Meetings.

Some two days before the meeting at which the foregoing paper was read, the authors decided to risk a series of experiments at speeds of 9 to 11 knots. These were carried out successfully, but the detailed results were too late for incorporation in the text of the paper. Generally speaking, the phenomena are the same as at lower speeds, but are more pronounced. The extreme rapidity with which the effects of interaction manifested themselves, and the difficulty of counteracting these effects by helm action, were in some cases so marked, that the authors would wish to modify the last paragraph of the paper, and to express the opinion that at such speeds the dangers of close proximity are great, no matter how well the helmsman may be prepared for any swerve.

DISCUSSION.

Mr. G. S. Baker (Member): My Lord and Gentlemen, these results are of special interest to us at the William Froude Tank, because some 18 months ago we made a series of experiments on suction with models. Unfortunately, I am not allowed to give you the results of any of these experiments, as they were done for those who still think the time is not ripe to make them public. But the experiments in deep water described in the paper, bear out in their general significance what we found with models in the deep water of the tank, and I think from that point of view alone the experiments were worth conducting. Many years ago, when William Froude propounded this idea of small scale experiments to represent what happened with big vessels, he had to go to large scale experiments to show that his method was reliable. Many people doubted whether suction experiments with models would apply to full-sized ships, and Mr. Thompson and Professor Gibson have done the thing that was necessary to prove this; they have shown, as we found, that suction is a real thing in deep water. The helm angles which they had to use in deep water at these speeds were not very great. Any man should be able to keep his ship out of danger if all the helm angle it was necessary to use was only 6° to 10°. Our experiments were mostly in shallow water, and there we found the danger to be much greater, and I think, if the authors could be persuaded to take their boats on to a flat shallow bank such as the Maplins, and repeat their experiments there, they would increase the value of the paper manyfold, because it is in shallow water that the danger of suction lies. The deep water suction is not a danger to the careful navigator, for you cannot call a helm angle of 6° to 10° dangerous. With regard to some of the points in the paper the authors draw attention to, the fact that if you allow the ship to come up under the influence of suction with a slight swerve the helm angle required is considerably increased. That is so, but it is not increased because of the suction. The suction influence is precisely the same, but in addition you have the inertia of the ship swinging it round, and only a little rudder to stop it, and if you allow a ship to start spinning like a top you cannot stop her for some time. But this is not suction; it is bad navigation to let a ship get a swerve on her when close to another vessel.

I would like to ask the authors whether in these experiments they did not find something that they have not mentioned in the paper, and which I found in our model experiments, in both deep and shallow water, that is, that in some positions the model was particularly sensitive to a little change in the conditions of running. With the models in such positions a small change in the fore and aft position, in the course, or in the helm angle, would cause the model to swerve or sheer outwards, where before the change it would move inwards towards the other vessel. We found this at first by the model running, not towards the other boat, but towards what represented the open sea. You will find that referred to by Mr. Taylor* in his paper on suction of some four years ago. He speaks there of the danger being not always of two vessels coming into collision with each other, but of one of the vessels being sent outboard and colliding with anything else that happened to be on that side of it. If you look through the discussion on Mr. Taylor's paper, you will find several cases given in which collisions have actually occurred through a vessel being, not sucked, but pushed away under the influence of another vessel.

Professor A. H. Gibson, D.Sc.: My Lord and Gentlemen, with regard to the points raised by Mr. Baker, it is very gratifying to know that the results of the large scale experiments agree so well with the results of the Teddington Model Tank experiments. I had the pleasure of seeing a number of those experiments, and can quite bear out what Mr. Baker says on that point. Mr. Baker raises the point that the helm angles which were found necessary to control the vessels in our experiments were comparatively small,

* Trans. Amer. Soc. Naval Arch., Vol. XVII. page 1.

but it must be remembered that the rudder which was in use in those particular experiments was very much larger proportionately than the rudder which would be applied to a large steamship, and, presumably, with a smaller rudder greater helm angles would be required. In our experience, when at a distance of half a length, we sometimes found an angle of as much as 17° necessary to maintain the course, and in one or two cases 35° was insufficient to keep the two boats apart. With reference to the point raised, that if the helmsman lets the attracted vessel swerve before he applies the helm, not suction, but bad navigation is to be blamed for the subsequent happenings, if a man is not anticipating the effect of suction, it is always possible that he allows his vessel to get a little out of control before he thinks it necessary to put his helm over. A vessel is always swinging from side to side a little, and if the beginning of a particular swing is due to suction, then the whole phenomenon must ultimately be put down to the effect of suction. We did find, as Mr. Baker intimated we should, that the behaviour of the small vessel depended very largely on its longitudinal position relatively to the larger vessel. In the majority of cases the smaller vessel was attracted, because we generally put it into a position suitable for attraction, but in some few experiments we put it into positions in which it should be, and was, repelled. A typical and well marked example of this is shown on Diagram 11 (Plate V.), in which the smaller vessel, slightly the faster, was brought up almost to the bow of the larger vessel. As it gradually drew ahead it got into a position in which it was repelled by the larger vessel, and was sheered out, as indicated in the diagram.

The CHAIRMAN (the Right Hon. Lord Inverclyde, Associate Member of Council) : I am sure, gentlemen, in your name I may thank Professor Gibson and Mr. Thompson for their extremely interesting paper.

EXPERIMENTAL DETERMINATION OF THE EFFECT OF INTERNAL LOOSE WATER UPON THE ROLLING OF A SHIP AMONGST A REGULAR SERIES OF WAVES.

By A. Cannon, Esq., Associate-Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913 ; Rear-Admiral the Most Hon. the Marquis of Bristol, M.V.O., President, in the Chair.]

§ 1. Before entering upon the subject of this paper, I shall ask you to allow me to outline briefly the sequence of events that led up to this experimental work, as it explains, in the clearest manner, the object of the study and its importance.

§ 2. Professor Sir John H. Biles, in the opening remarks of his Presidential Address to the Engineering Section of the British Association in 1911, said :—

" During recent years accidents have happened to ships, and they have mysteriously disappeared. The complete disappearance without leaving any trace has led to the assumption that the vessel has capsized. The circumstances of such cases obviously preclude the existence of any direct evidence. The only subjects of investigation can be (1) the condition of the ship prior to the accident, and (2) the probability that such a condition could be one which in any *known possible circumstances* could lead to disaster. The first is determinable by evidence in any particular case. The second involves a consideration of the whole question of the behaviour of ships at sea. What is the effect upon any given ship of a known series of waves ? What waves is a ship likely to meet ?

§ 3. " This subject has occupied the attention of scientific engineers, and it may be said to have been considered a solved problem. We have thought that, if a ship has a certain metacentric height and a certain range of positive stability, she is quite safe from the action of a series of waves of any kind which we know to exist. If, however, a known ship (and perhaps more than one) has these safety-ensuring qualities and mysteriously disappears, it may be desirable to review the grounds of our belief to see whether any *known possible combination* of circumstances may cause disaster."

§ 4. He then reviewed the whole subject of rolling of ships amongst waves and summarised his conclusions as follows :—

"(1) With wave-slopes of 3·6° the angles of maximum roll obtained in the *Revenge* with bilge keels may be taken as 22°.

"(2) This roll takes place when synchronism exists between the wave and the ship, when the wave is 910 ft. long and 18½ ft. high, and has a wave-slope of 3·6°.

"(3) Waves exist which are of this length, but which may have a height of 50 ft., and possibly more, and a wave-slope of 10°.

"(4) In such steeper waves we should expect to get much larger angles of roll.

"(5) Each ship has peculiarities of rolling due to its form, as well as to its lading and bilge keels, &c.

"(6) These peculiarities and the effect they have upon rolling, and the effect different waves will have upon the rolling of the ship, can best be studied experimentally."

§ 5. It had been the Professor's intention, however, to have placed before the British Association the results of an experimental study of the subject, but, unfortunately, an accident incapacitated him for some time, and it was impossible for him to get the apparatus ready.

§ 6. In December, 1911, I had the honour to be appointed by this Institution to the new Post Graduate Research Scholarship in Naval Architecture, and Professor Sir John H. Biles suggested that I should take up the experimental study of the rolling of ships amongst waves, and he placed the machine that he had designed for this purpose at my disposal. I accordingly submitted it as one part of a proposed scheme of research work, and the results of this study, so far as it has gone, constitute the subject of this paper.

§ 7. Before entering upon this experimental work I investigated the statical effects of loose water in ships. This formed the subject of a paper that I had the honour of submitting to this Institution last year. It was intended as a preliminary to the determination of the dynamical effects when rolling amongst waves.

§ 8. The machine that has been used for these experiments is in many ways unique, and is the only one of its kind in existence in this country. The principles underlying its design were pointed out to this Institution in 1900 by Colonel (then

Captain) Russo, R.I.N., in his paper on the Navipendulum; and also in 1902, when he gave the results of an experimental study of the rolling of ships amongst waves obtained by his apparatus.* As, however, there are several new features in connection with the machine at Glasgow, it may be as well to give a brief description of it and the theory underlying its design.

Theory and Construction of the Rolling Machine.

§ 9. A B and C D (Fig. 1) are two radii that revolve about A and C respectively, and are always parallel to each other. A C is a vertical line. A rod E is pivoted at B to A B, and can slide through a sleeve D that is pivoted to C D. Then, if the values of r_1 and r_2 are correctly chosen, and A B and C D are revolved at a particular uniform speed, B will represent the orbit circle of a trochoidal wave of height $2r_1$, and B D will always lie in the direction of the normal to the wave slope at B. Thus the resultant force on any particle at B will, at this speed of rotation, always act along D B.

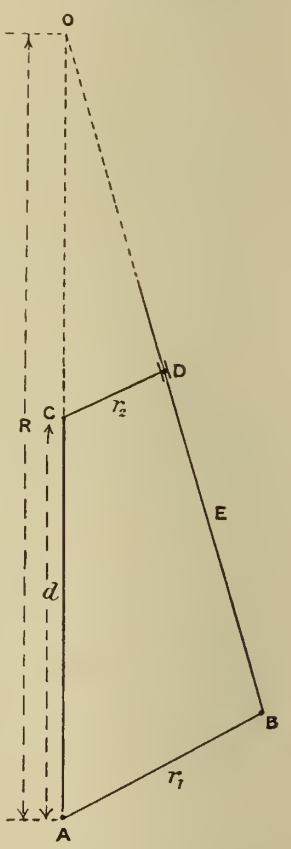


Fig. 1.

§ 10. The lengths of r_1 and r_2 and the speed of rotation may be obtained as follows:—

Let the machine represent a wave at a scale of $\frac{1}{n}$th the full size. Let l be the length of the actual wave and h its height. Then—

$$r_1 = \frac{h}{2n}$$

$$\frac{R}{r_1} = \frac{R - d}{r_2}$$

or—

$$R = \frac{r_1 d}{r_1 - r_2}$$

and, therefore, if r_1, r_2 and d are fixed quantities, R is constant; that is, A C and B D always meet in the same point O. Thus, if we make r_2 of such a length that R is equal to the radius of the rolling circle of a trochoid of length l, and rotate the machine at the appropriate revolutions, B D will always represent the normal to the wave surface; as it is a property of the trochoidal wave that the line joining any point of its surface to a point vertically above its orbit centre at a distance equal to the radius of the rolling circle is a normal to the surface at that point.

* Trans. I.N.A., Vol. XLII. page 30, and Vol. XLIV. page 76.

Now—

$$R = \frac{l}{2\pi n}$$

and—

$$r_2 = r_1\left(1 - \frac{d}{R}\right)$$

$$\therefore \quad r_2 = \frac{h}{2n}\left(1 - \frac{2\pi n d}{l}\right).$$

The period of a wave of length l is

$$\sqrt{\frac{2\pi l}{g}},$$

and, therefore, the period of the machine will be—

$$\sqrt{\frac{2\pi l}{gn}}.$$

So that, in order to represent a wave of length l and height h, we make—

$$r_1 = \frac{h}{2n},$$

$$r_2 = \frac{h}{2n}\left(1 - \frac{2\pi n d}{l}\right),$$

and rotate the machine at such a speed that its period is—

$$\sqrt{\frac{2\pi l}{gn}}$$

§ 11. The method of construction adopted for the machine can be seen in the detail drawings (Figs. 2 and 3, Plate IX.) and the general view (Plate VIII.). The main frame is of cast iron. The radii A B and C D (Fig. 1) are formed of slotted steel levers, so that their lengths may be set to any values desired. The machine is driven by an electric D. C. shunt-wound motor with a speed control in the armature circuit for the low speeds and in the shunt circuit for the higher speeds. This is directly coupled to a worm driving a worm-wheel attached to the lower lever. The upper lever is driven from the lower one by means of a bicycle chain that passes over two equal chain wheels, one of which is fixed to each lever.

§ 12. In order to secure uniform rotation and to prevent the machine from stopping too suddenly when the current is cut off, a heavy flywheel (see Fig. 2) is fixed between the motor and the worm. The speed of the machine can be varied from 15 to 40 revolutions per minute and the control resistances are very sensitive indeed, so that any desired rate of revolution can be obtained within the above limits.

§ 13. The machine was originally intended to be transportable, so that the various parts were made as light as was deemed advisable. It was found, however, that a good

deal of vibration was set up in the vertical standard when the motor was running. To remedy this a stay was taken from the top of the standard to an adjacent wall. It was also found that, owing to the great weight of the model—about 200 lbs.—some means of balancing this was necessary to ensure a uniform rotation. This necessitated making a continuation of the worm-wheel shaft and supporting its other end upon an independent bearing. The stay, flywheel, and counterbalance weight are shown in Plate VIII. and Plate IX. (Figs. 2 and 3). The whole stands upon a wooden base that is fixed to a specially prepared concrete floor.

The Model.

§ 14. It is well known that if a cylinder be made whose section is of the form of the locus of the projection of the centre of buoyancy of a ship upon a transverse plane as the ship is inclined, and the cylinder be so weighted that its centre of gravity is in a similar position to that of the ship, then if this cylinder be placed upon a horizontal plane the restoring couple at any inclination is directly proportional to the restoring couple acting upon the ship when at the same inclination. This principle has been made use of so often that it is unnecessary to demonstrate it here. It is also used in this case. Two steel plates are accurately made to the shape of the curve of buoyancy of the ship that it is desired to experiment upon, and the model is supported by them.

§ 15. Colonel Russo supports his Navipendulum upon curves that are involutes of the metacentric evolute and that are as near the centre of gravity of the ship as possible. This, however, necessitates very short curves, and hence increases the mechanical difficulty of making them accurate. Also, his trochoidal path passes through a point near the centre of gravity of the ship, whereas in this case it passes through the centre of buoyancy. This is referred to again later (see paragraph 21).

§ 16. The model itself consists of a wooden framework made to the shape of a section of a ship. Attached to it is a heavy gun-metal casting shown in Figs. 5 and 6 (Plate IX.), to which the supporting plates are fixed. These plates rest upon two steel rods (afterwards referred to as the roller paths), each having a V-shaped groove in its upper surface. The bottom of this groove is made of semi-circular section and the edges of the supporting plates are rounded, so that there is no rubbing between the sides of the grooves and the plates. Each roller path is supported upon two gun-metal wheels (Plate VIII.), mounted on ball bearings. These wheels are fitted to maintain the centre of gravity of ship in the position it will occupy relatively to the wave surface if its displacement in the wave remains constant.

§ 17. In order to obtain the correct rise and fall of G relative to the wave surface, these wheels should be cam-shaped. The shapes of the cams are difficult to determine.

Some have been designed for various shaped vessels, and are shown in Figs. 7, 8, and 9 (Plate IX.). A set was made for Fig. 7, but it was found that, as the model rolled, slipping took place between the cams and the roller paths. The wheels were then made circular. It was found that in some cases the cams would have re-entrant parts which would preclude their use.

§ 18. At the centre of gravity of the model a steel rod passes from front to back (Fig. 6) through two slots in the gun-metal casting that carries the buoyancy plates. The position of this rod can be so adjusted that the centre of gravity of the model is at its centre line. Ball bearings are fitted to each end of the rod, and these pass through slotted guides, so that the only possible motion of the centre of gravity is along a straight line. This line is parallel to the wave normal. When the machine is revolving at its correct speed the force between these ball bearings and the sides of the slots will be negligible and will not interfere with the rolling.

§ 19. In Colonel Russo's machine the curved supporting plates of his Navipendulum rest upon a rigid base plate, so that the instantaneous axis of rotation of the pendulum is the line joining the points of contact of the supporting plates and the base plate. Thus the locus of instantaneous axes of rotation is the cylindrical surface formed by the plates, whereas in this case it is the locus of Z, where G Z is the righting lever at any inclination (Fig. 10, Plate X.). This is similar to the upper half of the curves given in the paper read before this Institution in 1909 by Mr. A. W. Johns.* He showed that the locus should be a double-looped curve, intersecting at a point near the C.G. In Colonel Russo's arrangement the axis might be some considerable distance from the C.G.

§ 20. Double-bottom tanks, &c., can be fixed to the wooden framework. By means of slotted bars, shown in Fig. 4, weights can be attached to the model and moved vertically to get the correct position of the centre of gravity and horizontally to obtain the correct moment of inertia about the centre of gravity.

§ 21. The whole is so mounted upon the rolling machine that, when the machine is in an upright position, corresponding to a wave crest or trough, the point of contact of the supporting buoyancy plates and the roller paths supporting them, *i.e.*, the centre of buoyancy in the upright condition, is at the same level as the point B (Fig. 1). In this way the trochoidal wave produced is that which passes through the centre of buoyancy in the upright position, or, in other words, it is the effective wave acting upon the ship. This will be very slightly flatter than the surface wave.

§ 22. To obtain the correct C.G. of the model, the C.G. rod is set to its correct position, relatively to the C.B. The model is then supported by small wooden blocks

* Trans. I.N.A., Vol. LI. page 179.

placed in the slots beneath the ball bearings on the C.G. rod, and the weights adjusted vertically until the whole balances at any inclination. In this way the C.G. can be set very accurately.

§ 23. Having done this, the next step is to "wing" the weights until the correct moment of inertia of the model is obtained. The method adopted is to time the model for small angles of roll. Within the limits of isochronism the period of roll of a ship is given by—

$$\pi \sqrt{\frac{K^2}{g\,m}}$$

where K is the radius of gyration of the ship about a longitudinal axis through its C.G., m is the metacentric height and g the gravity acceleration. Now, K and m for the model depend upon the transverse scale of the model relatively to the ship, and are independent of the longitudinal scale. Calling the transverse scale $\frac{1}{n}$, as in the case of the rolling machine, we see that the period of roll of the model is equal to—

$$\pi \sqrt{\frac{K^2}{g\,m\,n}}$$

or $\frac{1}{\sqrt{n}}$ times that of the ship. The weights are then winged out or in until the period of roll at small angles is correct.

§ 24. The choice of the value of n in any case is governed by many considerations. In the model the limits of B G in the upright condition are 5 in. and 12 in. The limiting length of G Z, the lever of stability, is $4\frac{1}{4}$ in. If it is greater than this, the point of contact of the buoyancy plate and its supporting roller path would fall outside the wheels supporting the path. The required lengths and heights of waves to be experimented upon also determine the value of n. The greatest and least wave heights are the limits to which the lower lever of the machine can be set; these are 24 in. and 4 in. respectively, and are equal to one-half the maximum and minimum wave height. The length r_2 of the top lever has been shown to depend upon the wave height, length, value of n, and also the length d of Fig. 1 (page 78). In the machine d is 33 in. and the limits of r_2 are 3·8 in. and 21·75 in.

§ 25. Having fixed upon a suitable value of n subject to the above considerations the shape of the buoyancy plates can be determined, and the size and shape of any double-bottom tank, &c., with which it is desired to experiment can also be determined and its position with respect to the C.B. In the case of a tank the longitudinal scale is of importance, as it may be desired to experiment with a tank of known length. This scale is fixed by the relation between the weight w of the model and W of the ship. Let this scale be x. Then $W = x\,n^2\,w$. We know W, n, and w, so that x can be determined.

Resistance to Rolling.

§ 26. The researches of the late Dr. W. Froude led to the conclusion that the law of resistance to the rolling of a ship in still water at small angles was given by $d\theta = a\theta + b\theta^2$, where θ is the angle of roll from the vertical, and a and b are two coefficients that are constant for the same ship when under the same conditions of surface, displacement, and position of centre of gravity, but are different for different ships.

§ 27. From a consideration of the nature of the resistances to rolling that were likely to act upon a ship, he came to the conclusion that the a term in the equation for resistance was induced by a retarding couple whose moment was proportional to the angular velocity, and the b term was induced by a retarding couple of moment proportional to the square of the angular velocity. It was therefore considered that, if it were possible to apply to the model resistance couples varying as the angular velocity and the square of the angular velocity, together with means of varying each, independently of one another, the whole field of resistance could be dealt with.

§ 28. The first step, however, was to determine how much resistance was already acting as a result of pure mechanical friction and air resistance. At this time no suitable method had been devised of recording the angles of roll, so that the only means of determining the magnitude of the resistance was to heel the model to some known angle, and count the number of rolls before coming to rest. This was compared with all available data, and estimates were also made of the number of rolls that would be required under various resistances.

§ 29. These experiments indicated that the mechanical friction of the machine was too great, and steps were taken to reduce this as far as possible. The wheels supporting the roller paths (Plate VIII.) had originally plain bearings. These were made ball bearings. There was also a plain bearing at each end of the C.G. rod, where it passed through the guides (see paragraph 18). These were also altered to ball bearings. The grooves in the roller paths were considered to be too steep, and a new set of paths was made with a more open groove, and then the bottom of this groove was made semi-circular, and the edges of the buoyancy plates rounded, so that the contact was as near as possible along the edge (see paragraph 16).

§ 30. In this way it was found possible to diminish the friction very materially, and, although it could not be totally eliminated, it was judged to be reduced beyond what would be the amount necessary. This was afterwards verified when a method was devised of recording the angles of roll (see paragraph 52).

§ 31. The problem of arranging the necessary additional resistances in the manner desired has not yet been solved. Various methods have been suggested to accomplish

this. These have been correct in theory, but could not be practically applied. The initial idea was to drive from the model a small dynamo having a constant field. This would induce a torque varying directly as the angular velocity. Part of the current so generated was then passed into an electric magnet of constant air gap, fitted with a circular armature that was oscillated by the model. In this way a torque would be induced that varied as the square of the angular velocity. By resistances in the two circuits the amount of each torque could be independently varied. An apparatus was designed to carry out these ideas, but it was found that the mechanical friction involved was too great, and also that the small currents generated at the dynamo were too weak to produce any magnetic effect. They were all lost in overcoming the internal resistances.

§ 32. It is, of course, possible to have recourse to the device of Colonel Russo. He produces a resistance by causing a wooden pulley to rub against a stretched elastic belt. The shape of the pulley has to be determined for each case. So far, however, our efforts have been confined to arranging some means by which a variation of resistance can be easily accomplished. With respect to the experiments referred to in this paper, I am of the opinion that it is unnecessary to aim at any very faithful reproduction of the resistance. The reasons for this are stated later (see paragraph 52).

§ 33. In addition to the problem of how to produce the resistance, there is the question of the amount that is necessary. Very little data are available respecting the resistance to rolling of actual ships, and practically the whole of this refers to warships. And of what is available there is such a difference in the values of a and b (paragraph 26) for the different vessels that it is impossible to say from this what the resistance would be for a merchant type. Owing to the courtesy of Sir Archibald Denny we were supplied with the results of tank experiments upon models of Channel steamers. In Fig. 11 (Plate X.) is shown a curve* (No. 11) of declining angles for one case. This curve has been drawn to the full scope of the information available in this particular case. On the same diagram are shown curves that are drawn to the scope necessary for our purpose. It will be seen at once that it is practically impossible to tell what the resistance should be at, say, 30°. The most that we can do is to judge from known data whether it is very far from the actual amount. This does not detract from the value of the machine, as in all these matters it is comparative rather than actual results that we wish; and, indeed, the conditions of loading and state of the bottom of a vessel will materially alter her resistance to rolling, so that it is unnecessary to strain after great accuracy.

* This curve has been drawn in the diagram in such a position as to show its relation to curve No. 2, as this curve represents the still water resistance of the model during these experiments.

The Recording Gear.

§ 34. The problem of recording the angles of roll of the model was a difficult one, and attempts were made to solve it along various lines.

§ 35. The chief difficulty is that there is no point on the model that is fixed. In other words, the axis of rotation changes from time to time. The nearest point to a fixed one is the C.G.

§ 36. Colonel Russo uses the device of a third lever immediately beneath A B (Fig. 1), rotating parallel to it and of the same length. Then a rod joining the ends of the levers is always vertical. The recording gear is carried upon a table fixed to this rod, so that by means of a pen attached to the Navipendulum the angles of roll to the vertical are registered. The introduction of this third arm naturally makes the machine very high and much more massive. It is for this reason that it was not included in the Glasgow machine.

§ 37. The method ultimately devised is that shown in Fig. 12 (Plate X.). A gun-metal wheel A is fixed to the C.G. rod and oscillates with it. A steel lever N is attached to A by a ball bearing, and is constrained to pass between two guides P P, so that it always lies along the direction of the wave normal. At the end of this lever another wheel B is mounted, also on ball bearings. A cord R passes round these wheels. Then, as the model rolls, any point in R moves a distance proportional to the angle to the wave normal. This cord is attached to a small metal block D that forms one joint in a pantagraph as shown. The links C G, G D, and E F are equal, and F G is equal to D E, and F H to C F. Then the motion of H is an exact copy of that of D to a scale equal to $\frac{C H}{C D}$. The diameter of the wheels A and B and the length of the links have been so proportioned that the motion of H is $\frac{1}{10}$th in. for every degree the wheels are turned.

§ 38. A roll of paper is actuated by a small electric motor and cylinders as shown. The paper moves across a small metallic table, and the whole is mounted upon a wooden framework and attached to the rod E (Fig. 1), so that a pen attached to the pantagraph at H will move across the paper and produce a record of angles of roll relatively to the normal to the wave.

§ 39. As the model rolls, the C.G. rises and falls, so that the record must be corrected for this. To enable this to be done, a rod M is attached to N and moves with it. At its end a pen is fixed which registers the rise and fall of the C.G. The pens J and K are attached to electro magnets. One is connected to a clock, and registers half-seconds, with a break at every ten seconds. The other is connected to a circuit that is automatically closed when the machine is at its lowest position, *i.e.*, in the wave trough. From this

the inclinations to the vertical can be found by adding the inclinations of the normal to the inclinations of the model relatively to the normal. So far, however, it has not been possible to do this, as it is found that the point D has a certain amount of side play, no matter how tightly the cord is stretched. This is, of course, magnified in the record. It does not affect the angle registered, but alters its position relatively to the wave trough. Means are being devised to guide D so that it will always move along a straight line.

§ 40. As the object of this study is primarily the safety of ships, the angle to the wave normal is the most important. A vessel that always moves with the wave so that it is never inclined to the normal, may be said to be absolutely safe, although, perhaps, uncomfortable. The dangerous positions are those inclined to the wave surface, so that in investigating the question of the safety of ships at sea, we are really concerned with the rolling relatively to the wave normal.

§ 41. Referring again to Fig. 12 (Plate X)., a pen L is fixed, and draws a straight line that serves as a base line. A specimen of the record obtained is indicated in Fig. 13.

Experiment Results.

§ 42. As the only available information regarding resistance to rolling was for a Channel steamer, it was considered best to experiment at first upon this type.

§ 43. The dimensions of the vessel chosen for this purpose are 330 ft. by 42 ft. by 18 ft., service draught 11 ft. 9 in. moulded, and displacement at this draught 2,380 tons.

§ 44. From considerations already outlined the scale chosen for the transverse ratio of the ship to the model was 13·5. With this ratio, the following are the particulars of the ship and model :—

	Ship.	Model.	Ratio of Ship to Model.
	Ft.	In.	
G M	2·8	2·49	13·5
B M	10·6	9·42	13·5
K B	6·80	6·04	13·5
Period of single roll ...	6 secs.	1·63 secs.	$\sqrt{13·5}$

§ 45. The double bottom tank used was one that had been designed for a different vessel, and is shown in Plate VIII. It is fitted with five division plates, so that varying breadths of tank could be experimented upon. In these experiments only the two outer plates were kept in place, and water admitted into the space between them. The total

transverse breadth of this is 32 in. in the model, or 36 ft. in the ship. Thus the compartment stretches practically right across the ship.

§ 46. The height of the compartment is rather large, being 4 in. at the centre line and $3\frac{1}{2}$ in. at the wings of the model. These correspond to 4 ft. 6 in. and 3 ft. $11\frac{1}{4}$ in. respectively in the ship.

§ 47. The length of the compartment depends upon the longitudinal scale x (paragraph 25), which in turn depends upon the total weight of the model. The length of the model tank is $4\frac{3}{4}$ in.

§ 48. The model was tested upon waves of height equal to 15, 20, 25, 30, and 35 ft. For each height five different wave lengths were taken, viz., 600, 700, 800, 900, and 1,000 ft. One set of experiments was also carried out for a wave of 40 ft. height and 1,000 ft. long, but at this radius the machine was very difficult to balance, and there was a probability of overstraining it, so these experiments were not carried any farther.

§ 49. With each wave length and height the model was tried in six different conditions. First with no free water, and then by five successive additions of water, each of $2\frac{1}{2}$ lbs. weight. This alters the total weight of the model each time, and thus affects the value of the longitudinal scale x, and therefore the length of the tank. Owing to the free water surface the positions of stable equilibrium are inclined to the vertical. The following table shows the particulars of the ship corresponding to these conditions :—

Weight of water in model.	Weight of model.	Percentage of total displacement as free water.	Length of compartment in ship.	Inclination to vertical of stable equilibrium position.
Lbs.	Lbs.		Feet.	Degrees.
0	201	0		
$2\frac{1}{2}$	203·5	1·23	57·0	3·1
5	206	2·43	56·3	4·4
$7\frac{1}{2}$	208·5	3·60	55·6	5·4
10	211	4·75	55·0	5·9
$12\frac{1}{2}$	213·5	5·87	54·4	6·1

§ 50. In each case the C.G. of the model was kept fixed by moving the balance weights. With free water the C.G. is at the place it would be if the water were solid. Thus the effects produced are due merely to freedom of surface.

§ 51. The method of procedure was as follows :—

The magnetic brake that had been originally designed to develop the resistance to the rolling was mounted in position as shown in Plate VIII., and was connected to an external circuit. When a current is sent through this the model is held rigidly and cannot roll. The machine is then started, and when the proper revolutions are reached the brake is released when the model is in the trough of a wave. It was found that the brake blocks still touched their circular armature and thus produced a certain retarding couple. Experiments were carried out to determine the amount of this, and in how far it affected the rolling, and also at the same time to determine the effect of internal free water in quenching rolling in still water.

§ 52. Fig. 11 (Plate X.) shows a series of curves of declining angles that were obtained with the brake blocks, first in contact and then absolutely free, each with different amounts of free water. On the same diagram is shown the curve from the experiments of Sir Archibald Denny. It will be seen from these that the resistance with the brake blocks touching is nearer the required amount than when they are free. Indeed, if the Denny curve were extended to large angles, it would probably be steeper than this. But, apart from this question, as soon as a little free water is introduced into the model, the resistance is enormously increased, and when the amount of water is from 4 to 5 per cent. of the total displacement there is no appreciable difference between the curves of declining angles, whether the brakes are touching or not.

§ 53. As before stated, when the arms are set to their correct lengths, the machine only simulates the true wave motion when rotating at the correct speed. In order to ascertain the speed, the time to do about 20 revolutions is counted and compared with the correct time. The speed is then altered to suit and another record taken. The quantities required are the maximum angle reached and the amplitude of the forced oscillation. It was found that if several records were taken, as near around the correct period as possible, the results when plotted to a base of period of machine fall upon a regular curve, generally a straight line, and the values at the correct period can be obtained to a very fair degree of accuracy.

§ 54. All the records were subjected to a system of cross-fairing as shown in Figs. 14 to 19 inclusive (Plate XI.). In these diagrams the values of the wave heights and lengths are set out in two directions at right angles to each other, in perspective, and the angles reached are set out perpendicularly to these. This enables one to see at a glance the effect of wave height and length. Thus a plane section like A B C D in Fig. 14 shows the results for waves of the same height but of different lengths. While a section like E F G H shows the results for waves of the same length, but of different heights. With the longer waves of small heights, the oscillations were often of a double

nature, a maximum and minimum occurring alternately. They were not very great, and to facilitate fairing the curves were drawn through the mean value. These oscillations are of no importance in affecting the safety of the ship.

§ 55. The results have also been plotted as shown in Figs. 20 to 24 (Plate XI.). Each diagram is for waves of the same length, and each curve is for the same wave height, the abscissæ being the amount of free water in the ship and the ordinates the angles reached.

§ 56. Referring to Figs. 14 to 19 (Plate XI.), it will be noticed that, generally speaking, the shorter the wave and the greater the height, the larger is the angle reached. This is not always the case with respect to the wave height, especially with regard to the maximum angle reached, but it must be remembered that these angles are relative to the wave normal, and that, although this is smaller in some cases with an increase in height of wave, it is quite possible that the inclination to the vertical is not any less. This is borne out by the fact that the decrease of angle in any case is less than the increase of angle of the normal to the vertical, due to the increase of wave height. Another point to note is that the point of maximum angle to the normal is not necessarily one of maximum angle to the vertical.

§ 57. Another notable fact is that, as water is admitted, the superiority of the maximum over the forced oscillation dies away, and at 3·60 per cent. free water there is no difference. At this stage the ordinary cycles of roll no longer take place. The model rolls over to the angle of forced oscillation immediately, or else the roll gradually increases to a state of forced oscillation. In every case there was a very great effect noticeable upon these cycles, even with the smallest quantity of free water, indicating that a great deal of the energy was taken up in resisting the motion of the free water. Although, with internal free water, the positions of stable equilibrium in still water are inclined to the vertical, the forced oscillations when amongst waves were generally found to be equally inclined to the wave normal and not to the equilibrium position. In the longer and flatter waves, however, there was a slight tendency to oscillate about an inclined position, which was very near to the wave normal and never so much inclined to it as the still water equilibrium position. The total oscillation in these cases was always small. It is conceivable that in very long flat waves the oscillations may be about the equilibrium position, but in the shorter and steeper waves the internal water rushes across from side to side, and the rolling takes place about the wave normal.

§ 58. As the quantity of water increases, so the curves of angle of roll for constant wave length get steeper for an increase of wave height. With 5·87 per cent. of water on waves 600 ft. long and 35 ft. high, the motion was extremely violent; the water simply rushed in a foaming cascade from side to side, and the least disturbance only increased the effect.

§ 59. Synchronism should take place at a wave length of about 750 ft. long, but it will be seen that the maximum angles reached will be for a much shorter wave. Upon timing the model, however, it was found that the period for an oscillation of 36° was only 1·25 seconds, so that it is evident that the maximum angles are affected by the periods of roll for large angles, which one would expect to be the case.

§ 60. Referring to Figs. 20 to 24 (Plate XI.), it will be seen that, generally speaking, the addition of free water decreases the angle of roll. In the shortest and highest, and therefore steepest waves, however, the effect is to increase the angle of roll so that the forced oscillations are even greater than the maximum oscillations with no free water. This effect seems to be produced by the synchronism of the period of rush of water across the vessel, and the period of the wave. There will be a certain combination of wave length and height, *i.e.*, a certain wave steepness, that will produce just the critical period for the water to rush across. When this takes place, the angles of roll may be very large. Further, these angles are reached in a very few rolls and have been reached when starting from rest. It is, therefore, quite possible that dangerous angles may be reached in actual ships, where the vessel may have an initial heel and roll, although the resistance to roll may be very great.

§ 61. On the other hand, if the quantity of water be limited, the angles reached are always smaller. This points to the efficacy of anti-rolling tanks having small water capacity for the safety of ships.

§ 62. A great deal of work remains to be done on this very important subject. It is one of very great practical importance, especially in view of the growing tendency to adopt free water as a means of reducing rolling. It is proposed to carry out a similar series of experiments with anti-rolling tanks with a view to finding out experimentally the actual effects of such tanks upon the maximum angles of roll. Till such work is systematically done, we cannot be sure that in all circumstances anti-rolling tanks will reduce rolling. It is hoped that this paper will stimulate interest in the subject.

§ 63. In conclusion, I have to express my very great indebtedness to Professor Sir John H. Biles for having the machine made and giving me the use of it, and for the great interest he has taken in these experiments. I think this Institution is under a debt of gratitude to him for giving the machine for this purpose, as it has been a very costly apparatus, and for the time which he has spent in consultation with me in carrying out this work.

§ 64. My best thanks are also due to Sir Archibald Denny for his kindness in supplying me with some very valuable information respecting the rolling of ships, and also to Mr. S. J. Reed, of Messrs. J. H. Biles & Co., for his assistance in the mechanical arrangements of the apparatus.

After reading his paper Mr. Cannon gave a demonstration with the model.

For the Discussion on this paper, see page 108.

THE EFFECT OF WATER-CHAMBERS ON THE ROLLING OF SHIPS.

By Lloyd Woollard, Esq., R.C.N.C., Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 25, 1913; Rear-Admiral the Most Hon. the Marquis of Bristol, M.V.O., President, in the Chair.]

Of the various methods proposed for limiting the rolling of a ship in a seaway, the installation of a tank or channel partly filled with water has proved to some extent successful. Trials at sea have been carried out with a number of ships fitted with such tanks, commencing with H.M.S. *Inflexible* and including a number of recent passenger vessels; model experiments have also been made—to cite two instances, those by Sir Philip Watts, which are recorded in the Transactions of this Institution for 1885,* and by Major Nino Pecoraro, which are recorded in the Transactions of the Institute of Italian Naval and Mechanical Engineers.

The object of this paper is to investigate as far as possible on theoretical grounds the efficacy of water tanks as a means of reducing rolling at sea. It is believed that the results presented here may be of interest and value owing to the wide range of conditions that can be postulated; and that they may be of use in corroborating or explaining any peculiarities in the observations made either in ships at sea or in models. Details of the mathematical investigation are given in the Appendix to this paper (page 100).

Before stating the conclusions arrived at, it is considered desirable to outline the nature of the hypotheses, and to indicate the method by which the extent of the rolling under given conditions can be assessed.

In the theory of rolling founded by the late Dr. W. Froude (of which a brief recapitulation is here given), the wave is assumed simply harmonic, having a "virtual slope" expressed at any instant by $a \sin rt$, to which the ship constantly tends to incline. The apparent wave period is then $\frac{2\pi}{r}$, depending on the wave length and the speed and course of the ship. The maximum slope a depends on the ratio of the height to the length of the wave, and on the position of the ship relative to the wave; it does not generally exceed a few degrees among the waves encountered by large ships.

Assuming the resistance to rolling to vary as the first power of the angular velocity,

* Trans. I.N.A., Vol. XXVI. page 30.

and θ, the angle of heel, to be fairly small (say, up to 15°), the well-known differential equation—

$$\frac{d^2\theta}{dt^2} + 2K_1\frac{d\theta}{dt} + p^2\theta = p^2 a \sin rt \qquad (12)$$

is obtained, where p is 2π divided by the double period of unresisted roll of the ship.

The validity of the assumptions made has already been very fully treated, and it need not further be discussed here. It is of interest, however, to note that the linear law of resistance postulated above has been found to hold with a fair degree of exactitude up to about 6° of heel in a battleship of modern type, and that the value of the coefficient K_1 that is assumed throughout this investigation, viz., ·05, agrees fairly closely with that deduced for the ship. This resistance corresponds to an "a" coefficient of $\frac{\pi}{20}$ or ·157 in the decremental equation—

$$-\frac{d\Theta}{dn} = a\Theta.$$

On solving equation 12, θ is expressed as the sum of two terms—

(*a*) A "free" oscillation represented by

$$Be^{-K_1 t}\sin(t\sqrt{p^2 - K_1^2} + \beta),$$

which is damped, is arbitrary in amplitude and phase, and has a period equal to the natural resisted period of the ship; and

(*b*) A "forced" oscillation represented by—

$$p^2 a \cdot \frac{\sin\left(rt - \tan^{-1}\frac{2K_1 r}{p^2 - r^2}\right)}{\sqrt{(p^2 - r^2)^2 + 4K_1^2 r^2}}$$

that is undamped, has definite amplitude and phase, and is coperiodic with the wave.

Since any complex wave system may be resolved by a Fourier series into component simple waves of the type above considered, this result can be extended to cover the effect of a confused sea; each component gives a "forced" oscillation, and the resulting roll is obtained by adding together all these components. It is sufficient, therefore, to to consider the effect of a simple wave system on the ship.

When estimating the probable angle to which a ship will roll among waves of a particular period, both free and forced oscillations must be considered. From observations of rolling at sea it is known that a ship rolls irregularly with a period approximating to her own natural period; the free oscillation is not only present, but it dominates the period. The amplitude and phase of the free oscillation depend on the

initial conditions. If the assumption is made that the ship is originally at rest and upright on the crest or trough of the wave, the free and forced terms must then neutralise, so that the original amplitude of the "free" oscillation depends in such circumstances on the amplitude of the forced oscillation. The former is degraded (when resistance coefficient K_1 is ·05) to ·86 of itself each single roll, *i.e.*, in four single or two complete double rolls the free term is little more than one-half (·55) its original amount. The persistence of the free oscillation observed in practice is ascribed to irregularities in the wave system; a slight lengthening of one wave is sufficient to throw the system out of phase, and new free oscillations are thereby continually introduced.

It appears, therefore, that the most important factors affecting the angle of roll are—

(*a*) The magnitude of the forced oscillation, which governs also to some extent that of the free oscillation; and

(*b*) The coefficient of resistance K_1, which determines the rate of degradation of the free oscillation.

In dealing with the rather different problem of the maximum roll in a non-synchronous wave, Mr. R. E. Froude, in a paper read before this Institution in 1896,* combined these two factors, and obtained what was termed the "criterion amplitude of roll." This, however, is inapplicable when two free oscillations appear simultaneously (as will be found when rolling under the action of the water tanks); it is then necessary to retain the two criteria referred to above by plotting the magnitudes of the forced oscillation as a curve on a base of wave period $\frac{2\pi}{r}$, and by noting at certain points of the curve the corresponding resistance coefficient, which is variable when the effect of water chambers is included.

Curve I. (Fig. 2, Plate XII.) shows the forced oscillation thus plotted for a ship without water chambers. The dotted curve by it is the corresponding curve for an unresisted ship. It is evident that the curves are almost identical except over a narrow range of wave periods where these nearly coincide with the ship's period. It appears from this that the forced roll does not, when taken alone, provide an accurate indication of the actual rolling in all conditions; *e.g.*, among comparatively short or long waves its amplitude is unchanged by any additional resistance (due, for instance, to enlarged bilge keels). The necessity for taking some account of the resistance coefficient K_1 (which greatly affects the free component of the roll) is thus illustrated.

* Trans. I.N.A., Vol. XXXVII. page 293.

Effect of Water Chamber.

The assumptions made and the symbols used in dealing with the water chamber are given in the Appendix. The water is conceived to move along a tube having guide plates so that the effects of eddying and irregularity of motion may be neglected. The motion of the water relative to the ship is represented by the angle ϕ (see Fig. 1), so that if b is the distance of centre of tank at side from middle line of ship, $b\phi$ is the amount by which the water surface has moved from the mean position. A resistance coefficient K_2 is assumed for the tank similar to that for the ship: it is assumed that this can be increased or diminished as desired by throttling either the water in the channel or the air in the cross connecting pipe.

Four cases are taken—

(1) When the water in the tank is prevented from moving. This has already been considered.

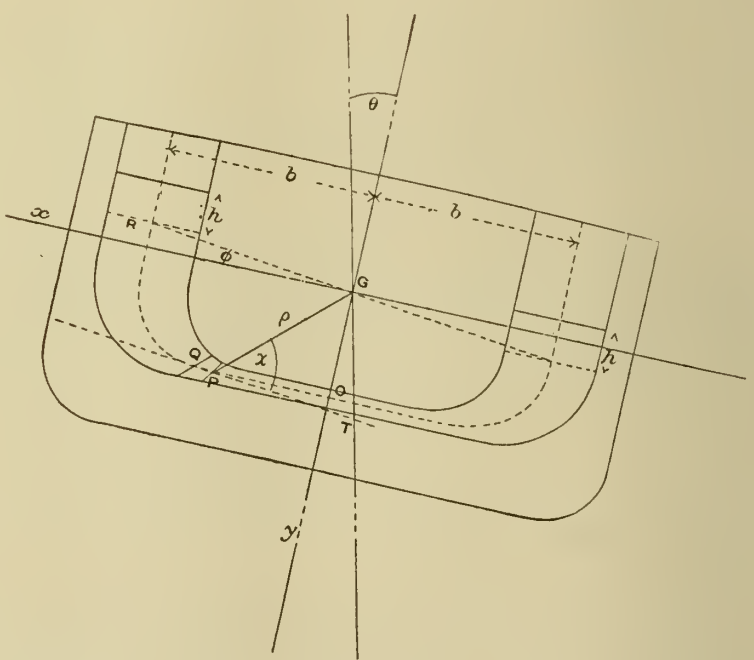


Fig. 1.

(2) When the ship is stationary and the water is allowed to oscillate in the tank. The equation shows that the motion is then a damped harmonic oscillation of period $\frac{2\pi}{q}$ where q is determined from a certain integral depending on the size and shape of the water chamber, which can be readily evaluated by graphical means. A moderate change of the water level in the tank or in the resistance due to throttling has no appreciable effect on the period of oscillation.

(3) When the tank is in operation but the motion of the water and of the ship are absolutely unresisted.

In still water, the ship and the water can describe simple harmonic oscillations in one of two periods determined by a quadratic equation (21). These are termed the principal oscillations of the system. In general both ship and water have oscillations arbitrarily compounded of these two principal oscillations, and are in consequence apparently irregular.

The periods of these principal oscillations, and, moreover, the complete solution of the motion under these conditions, depends upon two factors :—

(*a*) The fraction of the original metacentric height lost due to the movement of the water in the tank. This represents the power of the tank to affect by the statical position of the water the motion of the ship, and is denoted by μ. Its value can be calculated from equation (9) in the Appendix.

(*b*) The effect of the inertia of the water moving from side to side, denoted by a coefficient λ. This is determined by equation (5) in the Appendix; and involves the evaluation by a graphical process of a second definite integral. λ is generally fairly small; it is quite negligible when the horizontal portion of the tank passes close to the axis of oscillation, *i.e.*, to the centre of gravity of the ship. When this part of the tank is low in the ship λ is positive, and the inertia of the water opposes the statical effect expressed by the factor μ; on the other hand, when the tank is entirely high up, the statical effect is augmented by the inertia. The tank should, therefore, be as high as possible in the ship. It is assumed in the numerical results given that μ has the value of $\frac{1}{5}$, and that λ is negligible.

Among waves the oscillations of ship and water are each represented by two "free" oscillations of arbitrary amount and phase having the periods of the principal oscillations; and of one forced oscillation in the period of the wave. The latter is infinite when the wave synchronises with either of the principal oscillations, and zero when the wave period is about 10 per cent. greater than that of the tank alone. As is seen below, this result is greatly modified by the resistance of the ship and tank; but it points to the conclusion that, whatever the period of the ship may be, the rolling is a minimum when the wave period is slightly greater than the tank period. If it were possible, therefore, to modify readily, and to any extent required, the tank period, a large reduction in rolling could be ensured by rapid and continual adjustment of the apparatus.

In the most favourable condition, when the ship and wave are synchronous and the tank period about 10 per cent. less, the angle Φ (which defines the movement of the water in the ship) is equal to

$$\frac{a}{\mu\{1-\lambda(1-\mu)\}}, \text{ or } \frac{a}{\mu}$$

approximately. This determines the height necessary to allow the tank for an assumed wave angle ; for if h be this height,

$$\Phi = \frac{h}{2b},$$

and therefore—

$$h = \frac{2ba}{\mu} \left(\text{or } \frac{2ba}{\mu\{1 - \lambda(1-\mu)\}} \right).$$

Conversely for a fixed height of tank, the equation fixes the greatest wave angle a with which the tank can cope. When the resistances are taken into account it will be found, however, that the height h can be reduced considerably.

(4) When the tank is in operation, and ship and water are both subject to a resistance proportional to the velocity.

For simplicity, the effect of the inertia of the water, represented by the coefficient λ, is neglected entirely.

Both θ, the angle of heel, and ϕ, the angle subtended by the water surface, are composed of two free oscillations, damped, and of arbitrary amplitude and phase, superposed on a forced oscillation, undamped, and of definite amplitude and phase. The period of the latter is the period of the wave ; but the periods of the former are the periods of the principal oscillations of the system which, together with the damping coefficients, are found by solving equation (26) in the Appendix.

The effect of the free terms may first be briefly considered. They are two in number, either of which may preponderate according to the circumstances. If the motion grows to any large amount after a period of quiescence, one of the free terms must have been opposed in phase to the forced term, and nearly coperiodic with it. The two terms will then gradually come into phase, giving rise to fairly large angles of roll. If, on the other hand, the periods of the forced and of one free term are not approximately equal, it is impossible to get the state of no roll to endure for any length of time, for the relative phase of the two terms alters immediately.* If it be assumed, as appears reasonable, that periods of quiescence are experienced fairly frequently, it follows that the more important free term is that whose period is more approximately equal to that of the forced term.

* It is possible, of course, for three harmonic terms to produce a condition of no roll in an infinite variety of ways. However, for this condition to continue for some little time and then to give place to rolls of gradually increasing amplitude attaining a climax of heavy rolling, it is necessary for two of the terms to be nearly coperiodic and for the amplitude of the third to be negligible, as assumed.

It has been shown that the "K" damping coefficients limit the rolling (apart from their influence on the forced oscillation) by their effect on the free term. The values of K_3 and K_4, the damping coefficients of the free oscillations, are therefore marked on the curves of forced roll (II., III., IV., V., Fig. 2, Plate XII.) at points corresponding to their corresponding periods $\frac{2\pi}{n_3}$ and $\frac{2\pi}{n_4}$. From the preceding paragraph it is seen that over a range of periods near these points, these coefficients correspond to the damping of the more powerful "free" term, and they indicate approximately the degree of damping to be expected. In condition V., for instance, the coefficient is ·21 near the lower peak of the curve, and ·09 near the higher peak; the "free" component oscillations would, therefore, be rapidly damped near the former position, *i.e.*, among waves of comparatively short period, while their degradation would be much lower among longer waves.

The amplitude of the forced oscillation and the various coefficients have been calculated; the former is plotted in the form of curves on a base of $\frac{\text{wave period}}{\text{ship's period}}$ or $\frac{p}{r}$. The resistance of the water in the tank has been varied, but that of the ship is assumed constant ($K_1 = ·05$ giving a decremental "a" coefficient of ·157). In general the tank period $\frac{2\pi}{q}$ is ·9 times that of the ship, but in one case it has been diminished to about ·7 times that of the ship $\frac{2\pi}{p}$. The results are given in the table on the following page.

Curves II., III., IV., V. (Fig. 2, Plate XII.) indicate the effect of the tank. In II., where the tank resistance is low, the results already found for the condition without resistance are broadly confirmed. At the periods of the principal oscillations humps are formed, and the tank greatly enhances the rolling of the ship. When the ship and wave synchronise the rolling is enormously reduced. The damping coefficient is unaltered.

In curve III. the water in the tank is throttled, so that its resistance is increased five-fold. The humps are now less pronounced, and the gain at synchronism is correspondingly reduced. The damping coefficient is greatly increased—to about ·15, so that the free component of the roll would be greatly diminished.

In curve IV. the water is further throttled, so as to multiply its original resistance by 10. The curve now approaches curve I. with no tank, the humps previously situated at the principal periods have disappeared, and the hollow at synchronism with ship is replaced by a hump. The damping coefficient is again increased.

Of the three conditions represented by curves II., III., and IV., probably III. is

$\frac{q}{p}$ or $\frac{\text{ship's period}}{\text{tank period}}$	No tank	$\frac{1\cdot118}{\text{or } \sqrt{1\cdot25}}$	1·118	1·118	$\frac{1\cdot398}{\text{or } (1\cdot25)^{\frac{3}{2}}}$
$\frac{k_1}{p}$ or resistance coefficient of ship ...	·05	·05	·05	·05	·05
$\frac{k_2}{p}$ or $\frac{q}{p}$ times the resistance coefficient of tank	—	·05	·25	·5	·25
$\frac{k_3}{p}$ or damping coefficient for first principal oscillation	—	·05	·17	·276	·21
$\frac{k_4}{p}$ or damping coefficient for second principal oscillation	—	·05	·13	·274	·09
$\frac{n_3}{p}$ or period of ship ÷ period of first principal oscillation	—	1·28	1·244	1·08	1·48
$\frac{n_4}{p}$ or period of ship ÷ period of second principal oscillation	—	·78	·786	·854	·832
Number of curve in Figs. 2 and 3.	I	II	III	IV	V

the most favourable as regards the reduction of rolling over the whole extent of the curve. When considering the application to a large ship it is evident, however, that the periods of the waves actually encountered by such a ship are usually less than that of the ship. For purposes of comparison the lengths of the waves are shown below the base, corresponding to a ship of 14 seconds double period, moving perpendicularly to the direction of wave advance. Although the apparent wave period may be increased for the same wave length if the ship is moving partly in their direction of propagation, this effect is probably limited, since pitching rather than rolling would result if the vessel's length made too great an angle with the line of crests. It follows that in rare cases only can the wave period exceed or even equal that of the ship. The initial portions of the curves are therefore of greater practical importance than the remainder, and in all the cases considered the forced roll has been increased under short period wave impulses.

In order to reduce the rolling under these shorter waves, advantage can be taken of the fact that the forced roll is a minimum when the wave period is slightly greater than that of the tank. In curve V. the tank period is lowered to about 70 per cent. of the ship's period, so as to render the installation useful under waves of reduced period and length. The resistance in the tank is the same as in curve III. This condition is that which would also be obtained if the metacentric height were increased by about 45 per cent. As

compared with no tank, the forced roll is slightly increased for wave periods from ·55 to ·7 of the ship's period; but a big reduction results for ratios of from ·7 to 1·15. Actually the reduction of the free roll by the increased damping coefficient would widen the range of usefulness of the tank, probably sufficiently to render it of some service for all short wave periods.

The maximum angles Φ taken by the water relative to the ship are shown (in terms of the wave angle a) on Fig. 3 (Plate XII.), curves II., III., IV., V. Φ is actually limited by the height of the side of the tank, and the height required may be determined by the aid of these curves. If condition V. be regarded as the most important and useful, $\frac{\Phi}{a}$ does not exceed 2·5, except when the tank increases the roll. The tank need be designed, therefore, only to have a height $b\ \Phi$ or $2{\cdot}5\ b\ a$ (a in circular measure) above the mean water-level.

The phase differences between ship, wave, and tank have not been investigated in detail; but it is of interest to note that the modification in the rolling caused by the tankwater is due not only to the direct gravitational and inertia effects of the water, but also to the different attitude of the ship relative to the waves. For instance, in the condition when the tank is causing increased rolling, the waves are doing increased work on the ship, since energy is dissipated both in the tank and in the augmented resistance caused by the accelerated motion of the ship. The change in the work done on the ship by the waves is therefore of importance, and needs consideration when it is desired to ascertain the effect of any device whose object is the mitigation of the rolling of a ship.

Conclusions.*

Finally, the broad conclusions (which may, however, need some modification in the light of further experience) are as follows:—

(1) The tank should be placed as high as possible in the ship; the horizontal portion in particular should be near, or above, the ship's centre of gravity.

(2) The period of the tank depends on its shape, particularly the area of the constricted channel; it is almost independent of the amount of water contained, and of the degree to which the flow is resisted.

(3) The rolling will be a minimum when the period of the tank is rather less than the apparent period of the waves meeting the ship.

* Added after the Meetings.

(4) A tank having a period capable of adjustment would, therefore, if practicable, prove advantageous; failing this, the tank period should be made about 70 per cent. of the ship's period in large ships.

(5) A moderate resistance to the flow of water in the tank is on the whole favourable, *e.g.*, the flow can well be throttled until the amplitude of movement is reduced by one-half after each single oscillation.

(6) The tank is useless or disadvantageous among very short waves, and should then be put out of action.

APPENDIX.

It is assumed that the angular motion of the ship and the movement of the water surface are "small"; and that both the ship and the water are subjected to frictional or other resistance directly proportional to the velocity. The cross section of the water chamber, regarded as a tube, is assumed to be fairly small, and to vary sufficiently slowly to justify the method of "parallel sections." The tube is supposed to be symmetrical about the middle line, and to be vertical and of constant cross section at the sides.

Let θ = angle of heel of ship (*see* Fig. 1, page 94).
S = area of cross section of tube at any point.
A = constant value of S at sides near free surface of water.
s = arc P O measured along central line of tube.
u = velocity of water at P relative to tube.
p' = pressure at P.
w = density of water.
U = tangential velocity of tube itself at P.
F = tangential acceleration of tube itself at P.
z = height of P at any instant above a horizontal plane.
$f(s)$ = a frictional coefficient for water in tube at P.
G x, G y = perpendicular axes moving with the ship through G, the centre of gravity and assumed centre of oscillation.
h = height of mean water level in tube above G x.
R = a point situated a distance h below the momentary water level in tube.
ϕ = angle R G x.
χ = angle G P T, where P T is tangential to tube.
ρ = radius G P.
Θ = maximum value of θ.
Φ = maximum value of ϕ.
W = weight of ship including water.
m = metacentric height of ship assuming water immovable.
K = radius of gyration of ship about G.
$p = \sqrt{\frac{m \cdot g}{K}} = 2\pi \div$ double period of roll.
σ_1, K_1, = coefficients of resistance for ship when rolling.

Consider first the motion of the water relative to the tube. The tangential components of the forces in an element P Q of length ds are :—

$$\text{End pressures} = \frac{-\partial}{\partial s}(p' \, S) \, . \, ds$$

$$\text{Pressures from slant slides} = p' \, . \, dS$$

$$\text{Due to gravity} = -w \, . \, S \, . \, ds \, . \frac{\partial z}{\partial s}$$

$$\text{Due to friction} = -w \, . \, u \, . \, f(s) \, ds.$$

The rate of change of momentum consists of—

$$\frac{\partial}{\partial s}\left(\frac{w}{g} u \, . \, (u + U) \, S\right) . \, ds$$

as the water passes from P to Q ; and—

$$\frac{w}{g} S \left(\frac{\partial u}{\partial t} + F\right) . \, ds,$$

due to the acceleration of the water within P Q. Whence—

$$-\frac{\partial}{\partial s}(p' \, S) \, ds + p' \, . \, dS - w \, . \, S \, . \, ds \, . \frac{\partial z}{\partial s} - w \, . \, u \, . \, f(s) \, . \, ds$$

$$= \frac{\partial}{\partial s}\left\{\frac{w}{g} u (u + U) \, S\right\} ds + \frac{w}{g} S \left(\frac{\partial u}{\partial t} + F\right) ds,$$

or, dividing by $S \, . \, ds$,

$$\frac{w}{g}\left(\frac{\partial u}{\partial t} + F\right) + \frac{1}{S}\frac{\partial}{\partial s}\left\{\frac{w}{g} u \, (u + U) \, S\right\} + \frac{\partial p'}{\partial s} + w \frac{\partial z}{\partial s} + w \, . \, \frac{u}{S} \, . \, f(s) = 0 \qquad (1)$$

There is also the equation of continuity—

$$u \, S = \text{constant} = A \, b \frac{\partial \phi}{\partial t} ; \quad \text{or } u = \frac{A \, b}{S} \, . \, \frac{\partial \phi}{\partial t}. \qquad (2)$$

On substituting in (1), integrating throughout length of tube, the second term vanishes, since the tube is symmetrical about the middle line of the ship, and—

$$\frac{w}{g} b \frac{d^2 \phi}{d t^2} \int \frac{A}{S} ds + \frac{w}{g} \int F \, . \, ds + 2 w \, b \, (\phi + \theta) + w \frac{d \phi}{d t} A \, b \int \frac{f(s)}{S^2} ds = \text{constant}. \qquad (3)$$

In still water

$$F = \rho \frac{d^2 \theta}{d t^2} \, . \, \sin \chi.$$

The integrals can with but small error be taken between the limits of the *mean* free surfaces, so that they become constants. Put—

$$\int \frac{A}{S} \, . \, ds = \frac{2 g}{q^2} ; \quad \int \frac{\rho}{b} \sin \chi \, ds = \frac{2 g}{q^2} \lambda ; \quad \int \frac{f(s)}{S^2} ds = \frac{4 K_2}{q^2 A} \qquad (4), (5), (6)$$

where q has the dimensions of a frequency.

The equation (3) becomes—

$$\lambda\,\theta'' + q^2\,\theta + \phi'' + 2\,K_2\,\phi' + q^2\,\phi = \text{constant} = 0 \qquad (7)$$

since θ, ϕ, and their derivatives vanish together.

Again, consider the motion of the ship and water together. The system is subjected to the following couples :—

Weight of ship (including water in mean position)—

$$= -\,W\,m\,\theta.$$

Weight of water (additional due to displacement from mean position)—

$$= -\,2\,w\,A\,b^2\,\phi.$$

Resistance of ship—

$$= -\,\sigma\,\frac{d\,\theta}{d\,t}.$$

The rate of change of angular momentum is—

$$\frac{W}{g}\,K^2\,\frac{d^2\,\theta}{d\,t^2},$$

due to ship and water (in mean position), and

$$\frac{d}{d\,t}\int \frac{w}{g}.\,S\,u\,\rho\,.\,\sin\chi\,d\,s,$$

due to motion of water relative to ship.

The last expression becomes—

$$\frac{2\,w\,b^2\,\lambda\,A}{q^2}\,.\,\frac{d^2\,\phi}{d\,t^2} \text{ by equations (2) and (5).}$$

Hence—

$$\frac{W\,K^2}{g}\,\theta'' + \frac{2\,w\,b^2\,\lambda}{q^2}\,\phi'' + W\,m\,\theta + 2\,w\,A\,b^2\,\phi + \sigma\,\theta' = 0,$$

or, on reduction—

$$\theta'' + 2\,K_1\,\theta' + p^2\,\theta + \frac{\mu\,\lambda\,p^2}{q^2}\,\phi'' + \mu\,p^2\,\phi = 0 \qquad (8)$$

where K_1 is a frictional coefficient dependent on σ, and

$$\mu = \frac{2\,w\,A\,b^2}{W\,\overline{m}} \qquad (9)$$

The loss of metacentric height due to the free water surface is $\frac{2\,w\,A\,b^2}{W}$, or $m\,\mu$; so that μ is the fraction of the stability lost due to the free surface.

If the ship is among waves of apparent period $\frac{2\,\pi}{r}$, and of maximum virtual slope α, the right-hand side of equation (8) becomes $\frac{W\,m\,\alpha\,\sin\,r\,t}{W\,\frac{K^2}{g}}$, or $\alpha\,p^2\,\sin\,r\,t$. Equation (7) is also changed, since F is increased by $-\,g\,\alpha\,\sin\,r\,t\,\cos\psi$, where ψ is the slope of the tube to the horizontal. On integration and

reduction a term $a q^2 \sin r t$ is added to the right-hand side, so that finally the two following equations are obtained :—

$$\theta'' + 2 K_1 \theta' + p^2 \theta + \mu \lambda \frac{p^2}{q^2} \phi'' + \mu p^2 \phi = a p^2 \sin r t \qquad (10)$$

$$\lambda \theta'' \qquad + q^2 \theta + \phi'' + 2 K_2 \phi' + q^2 \phi = a q^2 \sin r t \qquad (11)$$

which enable the motion of the ship and of the water to be completely analysed.

Consider four cases :—

(I.) *Water in tank prevented from moving.*

On putting $\phi'' = \phi = 0$ in equation (10), the ordinary rolling equation

$$\theta'' + 2 K_1 \theta' + p^2 \theta = a p^2 \sin r t \qquad (12)$$

is obtained ; the solution is—

$$\theta = B e^{-K_1 t} \sin (t \sqrt{p^2 - K_1^2} + \beta) + p^2 a \cdot \frac{\sin \left(r t - \tan^{-1} \frac{2 K_1 r}{p^2 - r^2} \right)}{\sqrt{(p^2 - r^2)^2 + 4 K_1^2 r^2}} \qquad (13)$$

where B and β are arbitrary.

The first term is the damped free oscillation, and the second the undamped forc ed oscillation ; obviously the latter is a maximum when $p = r$, *i.e.*, when wave synchronises with ship. If K_1 is fairly small (as in all practical conditions) it is evident from equation (13) that the decremental coefficient a in the equation

$$- \frac{d\Theta}{dn} = a \Theta \qquad (14)$$

where n is the number of single rolls, is related to K_1 by

$$\pi K_1 = a p \qquad (15)$$

In the numerical examples $\frac{K_1}{p}$ is assumed to be ·05, so that a is ·157.

(II.) *Ship prevented from moving ; water allowed to oscillate in tube ; no waves.*

From equation (7)

$$\phi'' + 2 K_2 \phi' + q^2 \phi = 0 \qquad (16)$$

The motion of the water is, therefore, a damped harmonic oscillation of period approximately $\frac{2\pi}{q}$. Since—

$$q^2 = 2 g \Big/ \int \frac{A}{S} ds \quad . \quad . \quad . \quad (4)$$

the period is determined by the evaluation (graphical or by other approximate means) of this integral ; it depends only on the length and section of the water chamber.

(III.) *Ship among waves ; motion unresisted.*

Putting $K_1 = K_2 = 0$, equations (10) and (11) become—

$$\theta'' + p^2 \theta + \frac{\mu \lambda p^2}{q^2} \phi'' + \mu p^2 \phi = a p^2 \sin r t \qquad (17)$$

$$\lambda \theta'' + q^2 \theta + \qquad \phi'' + \quad q^2 \phi = a q^2 \sin r t \qquad (18)$$

whose solution is—

$$\theta = \mathrm{B}\,(n_1^2 - q^2)\sin(n_1 t + \beta) + \mathrm{C}\,(n_2^2 - q^2)\sin(n_2 t + \gamma) - \frac{a p^2 \{r^2(1-\mu\lambda) - q^2(1-\mu)\}}{(r^2 - n_1^2)(r^2 - n_2^2)}\sin r t \quad (19)$$

$$\phi = \mathrm{B}\,(q^2 - \lambda n_1^2)\sin(n_1 t + \beta) + \mathrm{C}\,(q^2 - \lambda n_2^2)\sin(n_2 t + \gamma) - \frac{a r^2 (q^2 - \lambda p^2)}{(r^2 - n_1^2)(r^2 - n_2^2)}\sin r t \quad (20)$$

where B, C, β, and γ are arbitrary constants, and n_1^2, n_2^2, are the roots of the equation—

$$(n^2 - q^2)(n^2 - p^2) - \frac{\mu p^2}{q^2}(q^2 - \lambda n^2)^2 = 0 \quad (21)$$

Clearly the forced oscillation of θ vanishes when $r = q\sqrt{1-\mu}$ (since $\mu\lambda$ is very small), and is infinite when $r = n_1$ or n_2. Taking μ to be $\frac{1}{5}$, it follows that the rolling is small when $r = \cdot 9\,q$, *i.e.*, when wave period is about 10 per cent. greater than that of tank. When $r = p = q\sqrt{1-\mu}$, and λ is negligible, equation (21) gives $\frac{n_1}{p} = \frac{5}{4}$, $\frac{n_2}{p} = \frac{3}{4}$ approximately; and at $\frac{4}{5}$ and $\frac{4}{3}$ of the ship's period the rolling becomes excessive.

When $r = p = q\sqrt{1-\mu}$ the amplitude of the forced term of equation (20) for ϕ becomes on reduction $\frac{a}{\mu} \cdot \frac{1}{1-\lambda(1-\mu)}$. When λ is very small this becomes $\frac{a}{\mu}$; the effect of λ is, when positive, to increase ϕ, or, for a limiting value of ϕ to reduce the wave angle a that the tank can contend with. Hence this hydrodynamical coefficient should be small or negative; *i.e.*, the tank should be as high as possible in the ship.

When necessary λ may be evaluated by equation (5),

$$\lambda = \frac{q^2}{2gb}\int \rho \cdot \sin\chi \cdot ds,$$

using a graphical or approximate method of integration.

(IV.) *Ship among waves: motion resisted.*

For simplicity assume λ negligible.

Equations (10) and (11) become—

$$\theta'' + 2\mathrm{K}_1\theta' + p^2\theta + \mu p^2\phi = a p^2 \sin r t \quad (22)$$

$$q^2\theta + \phi'' + 2\mathrm{K}_2\phi' + q^2\phi = a q^2 \sin r t \quad (23)$$

To find the free oscillations, make right-hand side zero and put,

$$\theta = \mathrm{B}\,e^{xt}, \quad \phi = \mathrm{C}\,e^{xt}.$$

On substitution,

$$\mathrm{B}(x^2 + 2\mathrm{K}_1 x + p^2) + \mathrm{C}\,\mu p^2 = 0 \quad (24)$$

$$\mathrm{B}\,q^2 + (\mathrm{C}\,x^2 + 2\mathrm{K}_2 x + q^2) = 0 \quad (25)$$

whence—

$$(x^2 + 2\mathrm{K}_1 x + p^2)(n^2 + 2\mathrm{K}_2 x + q^2) - \mu p^2 q^2 = 0 \quad (26)$$

The four roots are in all practical cases complex, so that—

$$r = -K_3 \pm n_3 i\,; \quad \text{or } -K_4 \pm n_4 i \ (i = \sqrt{-1}) \tag{27}$$

The free terms are, therefore—

$$\theta = L_1 e^{-K_3 t} \sin(n_3 t + \beta_1) + L_2 e^{-K_4 t} \sin(n_4 t + \beta_2) \tag{28}$$

$$\phi = L_1' e^{-K_3 t} \sin(n_3 t + \beta_1') + L_2' e^{-K_4 t} \sin(n_4 t + \beta_2') \tag{29}$$

where L_1, L_2, L_1', L_2', β_1, β_2, β_1', β_2' are constants, partly arbitrary, but connected by four equations.

Both θ and ϕ are, therefore, the sum of two damped harmonic oscillations, the damping coefficients and the periods being dependent on the roots of the biquadratic equation (26).

A convenient method of solving this equation is indicated in the numerical example given below. If $\frac{K_1}{p}$ and $\frac{K_2}{q}$ are fairly small fractions (up to about ·25) the following approximate formulæ (deduced from equation 26) are of service—

$$\left.\begin{aligned} 2K_3 &= K_1 + K_2 + (K_1 - K_2)\frac{p^2 - q^2}{R^2} \\ 2K_4 &= K_1 + K_2 - (K_1 - K_2)\frac{p^2 - q^2}{R^2} \end{aligned}\right\} \tag{30}$$

where—

$$R^4 = (p^2 - q^2)^2 + 4\mu p^2 q^2\,;$$

showing that K_3 and K_4 lie between K_1 and K_2, and are separated by $\frac{1}{2}(K_1 + K_2)$.

If in addition p and q are not nearly equal,

$$\left.\begin{aligned} 2n_3^2 &= p^2 + q^2 + R^2 \\ 2n_4^2 &= p^2 + q^2 - R^2 \end{aligned}\right\} \tag{31}$$

approximately, giving the periods of the principal oscillations.

To get the forced oscillation, equations 22 and 23 may be written—

$$(D^2 + 2K_1 D + p^2)\,\theta + \mu p^2 \phi = \alpha p^2 \sin rt \tag{32}$$

$$q^2 \theta + (D^2 + 2K_2 D + q^2)\,\phi = \alpha q^2 \sin rt \tag{33}$$

where D is the symbol of differentiation.

Hence—

$$\theta = \alpha \cdot \frac{-\mu p^2 q^2 + p^2 (D^2 + 2K_2 D + q^2)}{\Delta} \sin rt$$

$$= \text{(on making the ordinary substitution of } -r^2 \text{ for } D^2\text{)},$$

$$\alpha p^2 \cdot \frac{(1-\mu) q^2 - r^2 + 2K_2 D}{\Delta} \sin rt \tag{34}$$

$$\phi = \alpha \cdot \frac{-p^2 q^2 + q^2 (D^2 + 2K_1 D + p^2)}{\Delta} \sin rt$$

$$= a\,q^2 . \frac{2\,K_1\,D - r^2}{\Delta} \sin r\,t \qquad (35)$$

where—

$$\Delta = \begin{vmatrix} D^2 + 2\,K_1\,D + p^2 & \mu\,p^2 \\ q^2 & D^2 + 2\,K_2\,D + q^2 \end{vmatrix}$$

$$= \zeta + \eta\,D \qquad (36)$$

where—

$$\left.\begin{aligned} \zeta &= (p^2 - r^2)(q^2 - r^2) - 4\,K_1\,K_2\,r^2 - \mu\,p^2\,q^2 \\ \eta &= 2\,K_2\,(p^2 - r^2) + 2\,K_1\,(q^2 - r^2) \end{aligned}\right\} \qquad (37)$$

Hence—

$$\frac{\theta}{a\,p^2} = \frac{(1-\mu)\,q^2 - r^2 + 2\,K_2\,D}{\zeta + \eta\,D} \sin r\,t$$

$$= \frac{\{(1-\mu)\,q^2 - r^2 + 2\,K_2\,D\}(\zeta - \eta\,D)}{\zeta^2 - \eta^2\,D^2} \sin r\,t$$

$$= \frac{\zeta\{(1-\mu)\,q^2 - r^2\} + 2\,K_2\,\eta\,r^2}{\zeta^2 + \eta^2\,r^2} \sin r\,t + \frac{\{(1-\mu)\,q^2 - r^2\}\eta\,r - 2\,K_2\,\zeta\,r}{\zeta^2 + \eta^2\,r^2} \cos r\,t$$

By squaring and adding these coefficients the square of the amplitude Θ of θ is obtained.

Thus—

$$\left(\frac{\Theta}{a\,p^2}\right)^2 = \frac{\{(1-\mu)\,q^2 - r^2\}^2 + 4\,K_2^2\,r^2}{\zeta^2 + \eta^2\,r^2} \text{ on reduction,}$$

Or—

$$\Theta = a\,p^2 \sqrt{\frac{\{(1-\mu)\,q^2 - r^2\}^2 + 4\,K_2^2\,r^2}{\zeta^2 + \eta^2\,r^2}} \qquad (38)$$

On substituting for ζ and η from equations 37, $\zeta^2 + \eta^2\,r^2 =$ (on reduction)—

$$\{(p^2 - r^2)^2 + 4\,K_1^2\,r^2\}\{(q^2 - r^2) + 4\,K_2^2\,r^2\}$$

$$- 2\,\mu\,p^2\,q^2\{(p^2 - r^2)(q^2 - r^2) - 4\,K_1 K_2\,r^2\} + \mu^2\,p^4\,q^4 \qquad (39)$$

Reverting to equation (26), and its solution (27), it is evident that—

$$(r^2 + 2\,K_1\,r + p^2)(r^2 + 2\,K_2\,r + q^2) - \mu\,p^2\,q^2 \equiv (r^2 + n_3^2 + K_3^2 + 2\,K_3\,r)(r^2 + n_4^2 + K_4^2 + 2\,K_4\,r).$$

On changing r to $r\,i$ and then to $-r\,i$, and multiplying the equations together, the left-hand side becomes—

$$\{(p^2 - r^2 + 2\,i\,K_1\,r)(q^2 - r^2 + 2\,i\,K_2\,r) - \mu\,p^2\,q^2\}\{(p^2 - r^2 - 2\,i\,K_1\,r)(q^2 - r^2 - 2\,i\,K_2\,r) - \mu\,p^2\,q^2\}$$

which becomes on reduction the right-hand of equation (39) $= \zeta^2 + \eta^2\,r^2$.

Also the right-hand side of the identity is, on reduction—

$$\{(r + n_3)^2 + K_3^2\}\{(r - n_3)^2 + K_3^2\}\{(r + n_4)^2 + K_4^2\}\{(r - n_4)^2 + K_4^2\};$$

which expression is therefore equal to—

$$\zeta^2 + \eta^2\,r^2.$$

Hence, from equation (38),

$$\Theta = a\, p^2 \sqrt{\frac{\{(1-\mu)\, q^2 - r^2\}^2 + 4\, K_2^2\, r^2}{\{(r+n_3)^2 + K_3^2\}\ \{(r-n_3)^2 + K_3^2\}\ \{(r+n_4)^2 + K_4^2\}\ \{(r-n_4)^2 + K_4^2\}}} \qquad (40)$$

By a similar process the following equation for the amplitude of ϕ is obtained—

$$\Phi = a\, q^2 \sqrt{\frac{r^2\,(r^2 + 4\, K_1^2)}{\{r+n_3)^2 + K_3^2\}\ \{(r-n_3)^2 + K_3^2\}\ \{(r+n_4)^2 + K_4^2\}\ \{r-n_4)^2 + K_4^2\}}} \qquad (41)$$

From these equations the curves II., III., IV., and V. (Figs. 2 and 3, Plate XII.) have been obtained.

A specimen calculation is appended showing the method of obtaining n_3, n_4, K_3, and K_4 for curve III.—

$$\frac{q^2}{p^2} = 1{\cdot}25,\ \mu = \frac{1}{5},\ \frac{K_1}{p} = {\cdot}05,\ \frac{K_2}{p} = {\cdot}25.$$

Putting $p = 1$ the equation (26) becomes—

$$(x^2 + 0{\cdot}1\,x + 1)\,(x^2 + 0{\cdot}5\,x + 1{\cdot}25) - {\cdot}25 = 0,$$

or—

$$x^4 + {\cdot}6\,x^3 + 2{\cdot}3\,x^2 + {\cdot}625\,x + 1 = 0.$$

Put this equal to—

$$(x^2 + {\cdot}3\,x + K)^2 - (a\,x + b)^2,$$

or to—

$$x^4 + {\cdot}6\,x^3 + (2\,K + {\cdot}09 - a^2)\,x^2 + ({\cdot}6\,K - 2\,a\,b)\,x + K^2 - b^2.$$

Equating coefficients—

$$a^2 = 2\,K - 2{\cdot}21, \qquad 2\,a\,b = {\cdot}6\,K - {\cdot}625, \qquad b^2 = K^2 - 1\ ;$$

and on eliminating a and b—

$$K^3 - 1{\cdot}15\,K^2 - {\cdot}90625\,K + 1{\cdot}056171875 = 0.$$

Solving by Horner's method—

$$K = 1{\cdot}1056\ ;$$

whence—

$$a = {\cdot}041,\ b = {\cdot}471\ ;$$

and the equation becomes—

$$(x^2 + {\cdot}341\,x + 1{\cdot}577)\,(x^2 + {\cdot}259\,x + {\cdot}635) = 0.$$

giving—

$$K_3 = {\cdot}17,\ K_4 = {\cdot}13,\ n_3 = 1{\cdot}244,\ n_4 = {\cdot}786.$$

DISCUSSION ON THE TWO PRECEDING PAPERS.

Principal Sir Donald MacAlister, K.C.B., LL.D.: My Lord Bristol, Ladies and Gentlemen, I am glad to take this opportunity of saying how very willing and how very pleased we are to welcome the Institution of Naval Architects in this room and in this University. As you know, our department of Naval Architecture is associated with great names, the names of Elder, Elgar, Jenkins, and Biles; and I think you will gather from what will be brought before you to-day that the science of Naval Architecture is not likely to lack propagation within these walls. It is a long while since I worked at mathematics myself, but I have not forgotten the language, and I congratulate both the readers of these papers on having put to practical and immediately valuable use that which I studied only as a theoretical mental discipline.

Sir Philip Watts, K.C.B., F.R.S., D.Sc. (Vice-President): My Lord Bristol, Ladies and Gentlemen, I congratulate the Institution on these two very valuable papers. Mr. Cannon has achieved most valuable results, which open out quite a new sphere of interest and attraction, but the remarks I have to make refer more especially to Mr. Woollard's paper. This paper has involved an enormous amount of labour and calculation, and it will form a very important contribution to the literature of the subject. I am very pleased to find another member of the Royal Corps of Naval Constructors adding to the work done by several of the younger generation in the application of scientific knowledge to purposes of naval architecture, more particularly as this subject is one in which I have been personally interested for many years. It is a paper which we shall, most of us, require some time to consider. It is based on mathematical treatment which appears thorough and sound; but I think we must remember that in such treatment it is always a question whether the assumptions are judiciously chosen, so as to give truly significant results. I think there are many points in connection with it which experiment alone can clear up. If I may make a suggestion as to the arrangement of the paper, I should like to suggest that a concluding paragraph be added, giving a summary of the principal points dealt with, and that the nature of the assumptions made be more specifically stated. I think these additions would add greatly to the value of the paper.

Mr. Woollard refers quite rightly to the fact that in an irregular sea a ship rolls approximately in her natural period. This was observed and pointed out in connection with the earliest automatic rolling records, including those taken in connection with trial cruises of the *Devastation* and those of the old *Inflexible*. It means that the free oscillation is in general the dominant factor. This being the case, it follows that any rolling resisting device must have its period related to the period of the ship. Mr. Woollard deduces an important fact on page 95 in connection with rolling tanks, viz., that the rolling is a minimum when the tank period is slightly less than the wave period, and he suggests that a large increase in steadiness could be obtained by continually adjusting the tank period. Judging from Fig. 2 (Plate XII.), it appears that in a tank of small resistance, such as is illustrated by curve II., the action of the tank may increase the angle of roll; for example, at abscissa value ·8, the roll without the tank is about twice the wave slope, and with the tank five times the wave slope—a great increase—while the increase at abscissa value 1·3 is nearly as great. I think, since diminution of the period of the tank is efficacious, it would be interesting to obtain the curves similar to No. II., when the tank period is ·71, perhaps; however, Mr. Woollard could give us a general idea as to how this curve No. II. is affected by reducing the period.

The left-hand side of Fig. 2 is the more practical region, since it represents waves which the vessel is more likely to meet, and consequently the curve No. V. appears to be the most useful to

adopt, for the maximum ordinary rolling is very slightly increased by the tank, whereas with big rolling the tank is effective. This curve, however, involves a big increase of tank resistance, which Mr. Woollard suggests may be obtained by throttling. Another important point, deduced on page 95, relates to the inertia effect of the water in the tank; it is that for the best extinctive effect the horizontal connecting channel of the tank should be as high up as possible, somewhere about the centre of gravity of the vessel, as otherwise the inertia of the water will be in opposition to the loss of the righting couple caused by the free surface. The data given as regards the limits of oscillation required by a tank will be very serviceable to those concerned in fitting such devices.

The conclusions deduced in page 97 are obviously for a perfectly regular swell, which I think puts them more or less out of court for practical purposes, except as bearing on behaviour in regular and nearly coperiodic waves. Mr. Woollard states the principle of an irregular sea correctly in the first sentence of the last paragraph but one on page 92; but it seems to me that the last sentence, "It is sufficient, therefore, to consider the effect of a simple wave system on the ship," does not necessarily follow. I am quite sure that *technically* the paper is an exceedingly valuable piece of work, and will be of very great use to all those dealing with questions in connection with anti-rolling tanks.

Colonel G. Russo, R.I.N. (Member): My Lord President and Gentlemen, the subject of the paper which we have heard, and of the experimental demonstration which we have seen with so much satisfaction is one of very great practical importance. Professor Sir John Biles, to whom we all are much indebted from so many points of view, alluded to me yesterday as the father of this method of research, as my experimental instrument, the navipendulum, which I brought before the Institution of Naval Architects in London in 1902, was the first of this kind. If so, the father would not be the best person to speak about the merits of his offspring; but at this moment I am full of admiration both for the new apparatus which we have before us and for Mr. Cannon's research, by which he has attained considerable results in elucidating an important question. I am well acquainted with the defects of such instruments and with the corrections they require, and the painstaking care necessary in using them, and I therefore congratulate Mr. Cannon very heartily on his success.

I have had but a very short time to read the paper, which is of such a nature as to necessitate attentive study and meditation. The difficulties which we have to face in navipendular experiments on ships depend essentially on the following facts: first, the axis of oscillation in still water or upon waves does not pass exactly through the centre of gravity, though the instantaneous axis of rolling passes near that point. Secondly, the centre of gravity of the ship rolling on the wave does not follow an exactly circular path. However, the differences are of little consequence for the experiments when we have to deal with waves of sufficient length in proportion to the dimensions of the ship.

I cannot enter into detail about the point mentioned in paragraph 15 of Mr. Cannon's paper. I consider that with my apparatus a certain dipping motion, a rise and fall of the centre of gravity, arises from the fact that the curve of centres of buoyancy is not an arc of a circle, and that with the new instrument before us an identical dipping motion, exactly similar, takes place, since it has not been possible till now to adopt cam-shaped wheels for sustaining the roller paths, but I have reasons for thinking that no inconvenience at all arises from this from the point of view of practical results.

Another difficulty of the problem is the regulation of the resistance to be added to the navipendulum in order to approximate to the conditions of the actual ship. Mr. Cannon has found that the actual resistances existing in the apparatus have given in his case an effect nearly corresponding to that which was needed. I remember and would repeat here what Mr. R. E. Froude said about the system of resistance adopted by me: "The proof of the pudding is in the eating."

Mr. Cannon has succeeded in giving us the result of his experiments on the effect of loose water, tabulated in a manner which I admire for its simplicity and clearness. No less than 150 results of experiments are given in his diagrams, and he has arrived at conclusions which seem to me perfectly sound. Recently a somewhat similar research has been carried out in Italy by my colleague Major Pecoraro. He has experimented on a small scale with the Frahm anti-rolling tanks both with models in still water and with the navipendulum in the dockyard at Spezia. Another similar apparatus is in use at Genoa. An interesting paper has been read and published at the meetings of the Italian Society of Naval Architects in December last, describing Major Pecoraro's experiments, and in which there are many points of contact with Mr. Cannon's research. I see from paragraph 62 of the latter's paper that it is his intention to carry out experiments with anti-rolling tanks, and I think that a comparison of results thus independently obtained will be useful and important.

Mr. Lloyd Woollard follows the theoretical line of investigation, and he has given us an excellent paper, starting from the principles established by Mr. R. E. Froude. The phenomenon of the rolling of a ship among waves with free water in tanks is indeed a very complicated one, involving the combination of three oscillating motions, each of which is subject to complex laws, and I am inclined to think, as Sir John Biles pointed out, that experimental methods will probably lead to the best results, but nevertheless I find that Mr. Lloyd Woollard's paper is a most interesting one. His conception seems to me to be one which places a new weapon in the hands of investigators. Returning again to Mr. Cannon's paper, permit me, as a foreigner, although not a stranger to your country, to observe how well the Research Scholarship of the Institution has been placed, seeing that it has led to a piece of work of such importance. It certainly gives good promise for the future of research work in this country, where the application of the principle of mechanical similitude had its origin and its development, and where so many renowned names are connected with the study of the rolling of ships among waves.

Professor Sir JOHN H. BILES, LL.D., D.Sc. (Vice-President): My Lord President, Ladies and Gentlemen, I am particularly pleased to know that a meeting of the Institution of Naval Architects has been held in the class room of Naval Architecture in this University. We have had a great many meetings of naval architects here, though not as an Institution, and they have passed through this room in a steady procession into the Institution of Naval Architects, and into the work of naval architects in the world. It is a peculiar pleasure to me that I am able to be here to-day amongst the members of the Institution of Naval Architects.

The first thing that will strike those who know the facts is that the Institution has been the means of obtaining a scholarship which has enabled Mr. Cannon to devote more than a year to the investigation of this important problem in connection with this machine, and it is exceedingly doubtful whether this work could have been carried out in any other way. The work has been very great, and I am sure, now that I know what has been done, that my attempt to investigate the problem by means of a machine would have completely failed, because I could not possibly have devoted one-tenth of the time to it that Mr. Cannon has, and I am sure I could not have done it anything like so well. That is a benefit to naval architecture in general, and to the Institution of Naval Architects in particular, which is worthy of note. This method of investigation by research scholarship is very much better in its results than independent investigation by enthusiastic people whose time is largely taken up in finding their own bread and butter. The subject of the rolling of ships is one which has been considered to a large extent, but I venture to think it has not been exhausted to anything like the extent that some of us thought a few years ago. Some of us had begun to look upon the rolling of ships as a matter about which we knew almost all that was worth knowing, and it was only by a rude awakening in the form of accidents

which we could not explain, that our attention was forced to further consider the subject. That is what led me, in the Presidential address which I gave to the British Association two years ago, to review the whole subject of the rolling of ships.

There is one important thing that Mr. R. E. Froude investigated and discovered for us, which I think is of interest, and it is really the foundation of this investigation. You have heard Mr. Cannon and Mr. Woollard talk about "free oscillations" and "forced oscillations." What Mr. Froude found for us was that in non-synchronised waves, that is, waves whose periods did not coincide with the period of the ship, a uniform system of rolling was set up by these non-synchronised waves in the rolling of a ship. That explained to a great many of us why we did not get these violent rolls at sea that we expected from the theoretical investigation when the element of resistance was left out. The rolling of a ship freely in smooth water was found to be gradually reduced by the resistance, and rolling amongst waves is gradually increased by the impulse which each wave gives the ship. If the two are acting independently, if there be a free oscillation without any wave impulse, the resistance would inevitably lead to the ship coming to rest. The oscillation which the wave causes would inevitably cause the ship to upset under certain conditions, and it is a combination of these two—the degradation of rolling by resistance, and the impulse and increase of rolling by the wave action—which actually exists in a ship, and they very soon adjust themselves, so that the increase of rolling due to the impulse is just balanced by the decrease of rolling due to the resistance. In that way you get the free oscillation of the ship not gradually falling off in angle, and you get the wave impulse not gradually increasing the roll of the ship, but the two combining together and adjusting themselves at a certain point so that the rolling becomes uniform. That is a point Mr. Froude worked out very carefully. The interesting thing to us is that before that uniform state is reached there is a period when the free oscillation of the ship caused by the impulse of the wave is attempting to adjust itself to bring about the balance, and during that period before balance is reached you may get a larger angle of roll. We are not afraid of uniform rolling, for we know exactly what that is, but we do not know exactly what will happen before that uniform rolling sets in, and that is the reason why this machine was devised upon which Mr. Cannon has carried out his investigations. The method previously adopted of including resistance is the method of graphical integration introduced by Sir Philip Watts, I believe, and used a great deal by Mr. Froude. The method is exceedingly difficult of application. When you have carried it out for one case you have only got one result ; you can make a long investigation for one ship, starting under one set of conditions, one particular angle of beginning the roll, and one position of the wave, and after doing a great deal of work you can only get one result. If you have to extend that over all possible combinations of sizes of ships and waves, all forms of ships, all conditions of lading of ships, and if for each one you have a different angle of roll to start with, you can imagine the problem is unsolvable in any practical way. But you can make all these solutions simple by a machine of this kind. That is one of the reasons why such a machine was made.

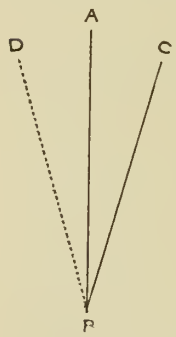
It is quite evident from what I have said already that this subject is one that cannot possibly be discussed with fulness in anything like the time at our disposal. The object of investigation was to find out whether any combination of circumstances could bring about an occasional large angle of roll, and Mr. Cannon has shown that under circumstances which are quite analogous to the every-day life of a ship you may, under certain conditions of wave which are quite practicable and probable conditions, with a certain amount of loose water in the ship, produce a dangerous condition of rolling. Now that is a very great encouragement to us to go on with this subject. We are only so far dealing with it in connection with the free water inside a ship. The investigation can also be carried out without any free water. We have not anything like covered the range of combination of wave and ship, and there is a great

field of investigation in this subject, but the encouraging thing, I think you will agree, is that so far we have found out some danger points which have to be avoided.

Now it is most interesting that in parallel with Mr. Cannon we have Mr. Woollard working along the purely theoretical line, and arriving also at results which point to possible conditions of danger, and I think when Mr. Cannon has had time to investigate from the same point of view as Mr. Woollard, and when Mr. Woollard has studied the results of Mr. Cannon's experiments, between them they will be able to completely determine the conditions which are likely to bring about dangerous results, so that we shall stand a better chance of avoiding them.

There is one point in Mr. Cannon's results upon which I should like to make one remark. The results speak for themselves; they are very clearly put both in the diagrams, and in the comments upon them that Mr. Cannon has made. In these results he has a condition of free water in which the ship has negative stability, but with a small inclination it has positive stability. That is a condition which is not uncommon at sea, and the rolling which Mr. Cannon shows takes place about the wave normal is easily explained, I think. One would suppose at first that a vessel which is inclined to, say, 6° before she starts to roll, would roll about that position of 6°, but there are two positions of 6° in a ship, one on each side of the middle line, and if she rolled about one of them she would roll about the other. The consequence is she rolls about the mean position. That is the rough explanation.

Now I have already, I think, told you in this Institution of experiences which I have had at sea in the rolling of ships which have a negative metacentric height, and which, therefore, have two conditions of rest about which they can oscillate. Supposing A B is the vertical, and the ship lies over to B C, she may oscillate about B C until she reaches the upright. When she reaches the upright she is free then either to come back to B C or to go over to the corresponding position of rest B D on the opposite side. If the impulse from outside is only sufficient to make her oscillate about B C she will not reach the upright; if it is sufficient to make her reach the upright, the moment she reaches it, instead of coming back she tumbles over to B D, and with the accumulation due to the roll she gets a larger angle; she has the accumulation of the whole of that roll. Then she comes back again, and if she passes the upright again she will tumble to the B C side, and if the angle she reaches is big enough she will oscillate from side to side about the middle. Presently the resistance will come in, and the vessel, instead of reaching the upright, will then roll about one side, B C or B D. So you get three series of roll; she rolls first about the port, then about the upright, and then about the starboard; with a little more she then rolls again about the upright, and with a little less she rolls about the port side; so with the present series of rolls, which are varying in character, they go through these phases. Mr. Cannon has pointed out from his experiments that when the wave is small the vessel tends to roll about the inclined position, which is a position of rest, but for all practical purposes, for the large angles he is considering and which are dangerous, she oscillates about the vertical.



I think we shall profit very much by the investigations of Mr. Woollard. Those who have been students here, and those who know what students are, know there is nothing a good student likes so much as a thick mathematical paper; he can get it up and can score in an examination with it. In that respect it is very much more valuable than mere practical results such as Mr. Cannon has got, which, however, are of value to those who have to conduct practical operations; but the good student will appreciate very highly, I am sure, Mr. Woollard's paper, and I thank both gentlemen for their valuable papers.

Mr. H. Wheatley Ridsdale (Member): My Lord President, Ladies and Gentlemen, I am sure the

members of the Institution will share with me the regret that Herr Frahm is not able to take part in this discussion; the fact is he is in America, and has not been able to see the papers. I have been asked by his technical associates to express on their behalf some of the reflections which are inspired by Mr. Woollard's paper. The very interesting investigations which Mr. Woollard has made in connection with anti-rolling tanks add considerably to the value of the practical and theoretical investigations brought before this Institution in 1911 by Herr Frahm* in connection with his now well-known invention. It would appear from a careful study of the paper that the only system of anti-rolling tanks which the author has taken into consideration is that due to and associated with the name of Herr Frahm, consisting in the use of water columns having a vertical movement, the period of oscillation of which is controlled by the dimensioning of the cross connecting channel, and up to a certain point by means of a valve in the corresponding air-connection. Although I know that the early work of Sir Philip Watts receives the recognition to which it is so richly entitled, and the name of the learned investigator Major Pecoraro, of the Royal Italian Navy, is quoted, I think the present author might usefully have made some specific reference to the work of, and to the paper read before this Institution by the originator of the system, the mathematics of which he sets out to develop. This system is, besides, the only one used in successful current practice. It is true that Herr Frahm, like nearly all other inventors, put practice before theory, and only attempted mathematical analysis after having attained a close approximation by model and full-size experiment. It is, of course, also well known that it is somewhat difficult to avoid errors when tackling such involved themes as, for instance, the rolling of ships at sea with anti-rolling tanks, solely from a theoretical point of view. I think it will, therefore, be of special interest to members to hear that results such as we have to-day had presented to us, and which have been arrived at with the aid of theory alone, are confirmed by the results obtained in a practical manner, and in actual ships.

It will be sufficient to refer to one or two of the more important points. Firstly, the author's remarks confirm the observation already made use of by Herr Frahm, viz., that a ship rolling at sea has her own natural period. That Herr Frahm was correct in basing the design of his tank on this fact has therefore received additional corroboration. It has also been observed, not only in the case of those installations which are already in use, but also during very careful model experiments, that a difference of period between that of the tank and that of the ship only appreciably affects the efficiency of the tank when its period is slower than that of the ship, the efficiency being then somewhat rapidly diminished. If, on the other hand, the period of the tank is somewhat quicker than that of the ship, the efficiency is increased, and most so when the period of the tank is by a certain percentage quicker than that of the ship, the exact percentage depending on practical and model trials. Beyond this percentage the damping effect slowly commences to fall off. From this it will be observed that the tank will always be effective if the period of the water column is calculated on a basis of the ship's own quickest natural period, and for all slower periods which may occur in the case of this particular ship the reduction in efficiency will not be appreciable. It is, of course, unnecessary to remark that no ship has at all times the same invariable period, but this varies within definite limits according to the condition of loading. It is, however, generally the case that no arrangement need be provided for regulating the period of the water in the tank, thus simplifying greatly its practical working. Up to a certain point the regulating of the period can be effected with the aid of the air valve. It has been proved by actual observation at sea to be quite unnecessary to have an exact adjustment between the period of the tank and that of the waves, such as the author indicates in his paper. Again, when the wave period

* Trans. I.N.A., Vol. LIII. (Part I.), page 183.

varies greatly (*e.g.*, by 40 to 50 per cent.) from the point of synchronism, no excessive rolling can occur. This fact is clearly brought out in Fig. 2 of the paper before us, and it will be seen from it that the amplitude of the ship's roll, without the tank, is only about one-fifth of the maximum when the wave period is 40 to 50 per cent. greater than that of the ship, and about 1/25th of the maximum when the wave period is 40 to 50 per cent. less than the ship's natural period, when, of course, the maximum inclination occurs. It is, therefore, obvious that in these two conditions the damping effect of an anti-rolling device is no longer required.

Mr. Woollard's investigations deal more particularly with the relationship between the period of the ship and that of the tank, which he assumes to be constant, and that the tank period is 10 per cent. quicker than that of the ship, whilst on the other hand the wave period varies. The results of these investigations are given in Fig. 2 (Plate XII.), and I may say that they resemble very closely those arrived at by Herr Frahm from model experiments with an apparatus generally similar to the Russo navipendulum, but much simpler. The results of the experiments with models and real ships fitted with tanks have only been published to a small extent. It is hoped that in the near future they will be published *in extenso*. There is a very close agreement between Herr Frahm's results and those of Major Pecoraro's model experiments. When lately in Spezia I had an opportunity of seeing the Russo navipendulum with Frahm anti-rolling tanks fitted on it, and it was very evident how admirably adapted machines of what I may call the Russo-Biles type are, not only for free water experimental demonstration, such as we have witnessed to-day, but also for investigating the possibly more immediately interesting question of anti-rolling tanks. There is, finally, therefore, in these results as set forth by Mr. Woollard and Major Pecoraro respectively, a happy example of agreement not only with the figures of the inventor of the Frahm tank, but also between pure theory and practical results realised at sea.

Sir Archibald Denny, Bart., LL.D. (Vice-President): My Lord President, Ladies, and Gentlemen, before Mr. Cannon and Mr. Woollard reply, may I make a remark with regard to this machine. I saw the ladies, when the experiments were going on, looking exceedingly alarmed. I suppose they wondered if it was at all possible that they had ever seen anything like that motion on a ship. They have, no doubt, but the motion was slower, and the waves were probably not quite so steep. As to the danger point that Sir John Biles referred to, he does not, I am sure, wish us to think that the danger is recurring every moment of the day, or every day of the week, or even every year. We do not want to alarm the public. There are some scientific medical men always looking for microbes, and when we hear of these microbes being discovered, we are alarmed, and wonder that we live at all; still, we do go on, and so ships will go on living at sea.

As to the statement of Mr. Woollard in the last paragraph on page 92, "From observations of rolling at sea it is known that a ship rolls regularly with a period approximating to her own natural period," I was of that opinion also, but my firm have been through experiments with anti-rolling devices, one of them with Professor Crémieu's apparatus. We tried that on a small steamer about 195 ft. long, and we sent along with Mr. Crémieu a skilled assistant to observe the behaviour of that ship. The inventor himself was *hors de combat* in a very short time: he could not stand it at all. The other man was proof against sea-sickness, and he assured me the period changed at sea altogether, and greatly. I will have that matter looked up, and will communicate with Mr. Cannon about it.

Sir Philip Watts: Was the rolling automatically recorded?

Sir Archibald Denny: No; it was done as best he could observe it. He said it was apparent even without observation.

Sir Philip Watts : I have some doubt about that, as I have had a great many examples of that sort of thing.

Sir Archibald Denny : We have tried experiments with the Crémieu and another apparatus which is well known, both on small vessels. With the other apparatus the vessel was still smaller, about 80 ft. long, and in still water the effect was very great : it was like a blow. When we sent the vessel to sea it was not successful, and we came to the conclusion it was because the vessel was too small, and that every little wave knocked her about, and she did not roll uniformly. Therefore, as the apparatus depends upon uniform rolling for its effect, it failed.

I have been through the Royal Naval College, as Mr. Woollard has, and I appreciate very highly the work he has done. It is exceedingly gratifying to find that the race of scientific naval architects is not in any way dying out, and that the Royal Naval College and Glasgow University and other Universities are producing able men to take the place of the old boys.

Mr. Lloyd Woollard (Member) : My Lord, Ladies and Gentlemen, before replying, I should like to congratulate Mr. Cannon on his very interesting paper, and to express the hope that he will be enabled to continue his experiments with a view to elucidating some of the very difficult problems associated with the rolling of ships. His results may be compared with those given in my paper, and on allowing for the fact that his ship has a single period of 6 seconds, and that of mine is 7 seconds, it is seen that his work deals with waves which, on Fig. 2 of my paper (Plate XII.), would have lengths varying between 800 and 1,350 ft., *i.e.*, for periods approximately synchronising with the ship's period. If, as Mr. Cannon suggests in paragraph 59, the natural period of his ship be reduced, their equivalent lengths would be from 1,400 to 2,300 ft. I hope Mr. Cannon will be able to extend his experiments, particularly among shorter waves, as it will be interesting to see how far the humps in the curves of forced oscillation are found to exist experimentally. I think it would also be interesting if he would take the case which is perhaps a little more practical than the one he is taking, that is to say, a ship with some initial stability. I am not very well acquainted with what has been done recently on actual ships with rolling tanks, but I do not think, as a general rule, that they fit a tank with such a free surface that the stability vanishes entirely. It would be more interesting from this point of view if he reduced the extent of the free surface, and if he took a ship whose period did not alter so much with the range of oscillation.

Sir John Biles : That is a practical form, you know. It is not like a warship form, but it is the form of many a ship actually running now.

Mr. Lloyd Woollard : It has a high freeboard, and this introduces a complexity which I should think it would be as well to avoid if possible in the initial experiments.

I wish to thank Sir Philip Watts for his remarks, and for the interest that he has shown in regard to this paper. In certain respects the assumptions made therein are necessarily at variance with known facts, and they include the simplifications introduced by Froude when the theory of rolling was first propounded. The conclusions were not elaborated in detail, for it was considered that they need confirmation by experiment and experience at sea. The paper is intended primarily to indicate fields for useful research on the lines carried out here by Mr. Cannon, and in Italy with the navipendulum.

If the resistance in curve V (Fig. 2) be reduced to that in curve II, I think that the effect would be to depress the hollow at abscissa ·8 and to raise considerably the hump at ·7. Rolling would probably be increased among short waves, but it would further diminish among all waves more than 600 ft. in length. The justification for the simple wake assumed lies in the fact that all the equations are linear, so that the angle of roll on a wave compounded of numerous simple waves is simply the sum (allowing for phase) of the

angles obtained with each separate wave. The linearity of the equations is, in turn, dependent on the assumption that the resistance varies simply as the angular velocity. This appears to be fairly correct in some ships, though in others it is only very roughly true; in such cases it provides, unfortunately, a limit to which any theory of rolling can be applied to a ship. Mr. Ridsdale's remarks can be replied to more satisfactorily in writing, but I am pleased to know that the results in ships and models to which he has alluded broadly confirm those here indicated. In conclusion, I wish to thank the various speakers for the encouraging reception they have given to this paper.

* Some of the points raised by Mr. Ridsdale appear to be rather outside the scope of this paper, which does not attempt to deal with the type or shape of tank, or with the practical points associated with their use. As far as is known, the model experiments cited are the most complete and systematic series of experiments whose results have hitherto been published. Sir Philip Watts pointed out at the close of Herr Frahm's paper, read at the 1911 meetings of this Institution, that his own work covered the form of tank re-invented by Herr Frahm, and Sir William White and Sir John Biles confirmed his statement; but the work done by Herr Frahm, and the interesting records he obtained, both from models and from ships at sea, are willingly acknowledged. As stated by Mr. Ridsdale, the mathematics apply more directly to a tank having a completely submerged horizontal channel; but an examination of the problem shows that the results hold good (except that for the period of the tank) for an open tank, though probably less exactly than for a closed tank. This conclusion is supported by some experiments made by the author with a model apparatus, representing the conditions obtained at sea amongst waves, though hardly so accurately as does the navipendulum. These were carried out at the Engineering Laboratory of the Royal Naval College, Greenwich, through the courtesy of Professor B. Henderson, D.Sc., to whom, together with his assistant, Mr. F. Bacon, M.A., the author wishes to acknowledge his indebtedness for the whole of the apparatus used. Both closed and open tanks were tried, and it appeared therefrom that the type of tank did not make a large difference to the results obtained. In all cases a fairly close approximation to the curves in Fig. 2 was arrived at.

Mr. A. Cannon (Associate-Member): My Lord President, Ladies, and Gentlemen, I must thank you all very much for your kind reception of my paper. With respect to Colonel Russo's remarks, he referred to the difficulties of conducting experiments with his machine, and I appreciate those difficulties. It is wonderful how much time is spent in trying to arrange small details, and I think that sometimes needless time is lost in trying to get the machine too perfect. With respect to dipping oscillations and the fitting of the cams, we have some cams here that were actually made, but when they were in place it was found that slipping occurred between the roller path and the cam surface, but no doubt that could be overcome by some means of roughening the surfaces. Dr. W. Froude investigated the question of dipping oscillations theoretically. He took a form which he called a peg-top section, in which the dipping oscillations would be a maximum, and came to the conclusion that even in that extreme form they might be neglected, in so far as they affected the angles of roll.

With regard to Major Pecoraro's experiments, as Colonel Russo has indicated, he experimented upon tanks of Frahm's form in which there is very little free surface, and the damping is produced by the vertical oscillation of the water. In our experiments there was merely a tank of the general double-bottom section, in which there was considerable free surface, and in which the water simply rushed from one side to the other. Why that form was taken first was because there are very few ships with Frahm's anti-rolling tanks, but there are a great many ships with double-bottom tanks, and the question was,

* Added after the Meetings.

if there was any leakage, or if either of the tanks was not sufficiently filled when it was necessary to have water ballast, to find if, under those conditions, a large angle of roll could be produced amongst waves. As stated in the paper, it is our intention to deal with anti-rolling tanks as well.

Mr. Woollard referred to the question of the range of his paper compared with the range of mine. As far as these experimental results are concerned, as I attempted to indicate in the paper, the question of range is concerned with the question of the dimensions of the machine, and for this particular vessel, for the value of what I have called n, the ratio of the transverse dimensions of the ship to the model, the range that has been taken is practically the full scope of the machine. I attempted to go a little beyond what the machine was capable of standing, but was afraid to do any more experiments in case there should be a fracture ; of course, it is possible to conduct some experiments upon the same form over other ranges by simply altering the value of this transverse ratio. That would necessitate making a new curve of supporting plates, and possibly a new tank as well. It is very interesting to find that Mr. Woollard has found from theoretical considerations with the Frahm tank the same results as were obtained in the other shape of tank with a free surface. Mr. Woollard said it would be interesting to get results from a ship with stable equilibrium even with a free surface. Of course, that could be done by putting in more division plates, but, as I said, my idea was to find if the machine really did work, and it was only possible to take the simplest case first. Even these experiments have taken up a good deal of time. If I had attempted also to discover what the effect would be of putting in division plates, the paper would not have been written to-day. That question would be of great interest, and no doubt we will investigate it later.

The PRESIDENT : Ladies and Gentlemen, I think, in the first place, we should give a hearty vote of thanks to Mr. Cannon and to Mr. Woollard for their papers. I should like also to take this opportunity of saying how glad the Institution of Naval Architects is to hold a meeting in this commodious and spacious room, and to acknowledge our indebtedness to Sir Donald McAlister and the Department of Naval Architecture in this University. I should also like to take this opportunity of thanking Sir John Biles for presenting this model to Mr. Cannon for the purposes of his investigation. You are now going along to other places of great interest, and I do not wish to detain you longer, so I will ask you to give a hearty vote of thanks to Mr. Cannon and Mr. Woollard for their valuable papers.

ON THE CRITERION FOR THE OCCURRENCE OF CAVITATION.

By Professor Dr. L. GÜMBEL, Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913; Rear-Admiral the Most Hon. the Marquis of BRISTOL, M.V.O., President, in the Chair.]

As a result of his experiences on board the torpedo-boat destroyer *Daring*, Mr. Sydney W. Barnaby* suggests as the criterion for the occurrence of cavitation in screw propellers that the pressure must not exceed about 11 lbs. per square inch of projected blade surface. Mr. D. W. Taylor, however, is of the opinion that this criterion is by no means absolute, since instances of cavitation in screw propellers have been observed when such surface pressure only amounted to from 4 to 5 lbs. per square inch. Mr. Taylor considers that this setting in of cavitation depends a great deal more on the following:—

(1) The speed at which the propeller blade moves through the water.

(2) The shape of the blade.

The researches described below are intended to show that this occurrence of cavitation depends upon:—

(1) The absolute pitch of the cross section of the blade.

(2) The angle ε between the tangent of the following edge and the face of the blade in the cross section of the blade (see Fig. 3, Plate XIII.).

(3) The amount of the slip speed as a function of the vessel's speed.

(4) The temperature of the water, the degree of its saturation with air, the barometric pressure, and the depth of the blade under the surface of the water, and

(5) *Not upon the mean pressure per unit blade area or upon the tip speed.*

When a screw propeller blade, having a pitch p, a diameter d, and n revolutions per second, moves through the water with a speed of advance of V feet per second (speed of advance of propeller being speed of ship minus speed of wake = V), then the water on the influx or suction side of the blade can only be in contact with the latter as long as the velocity of the water, obtainable by the over-pressure of the atmosphere, and by the pressure of the column of liquid, as against the tension of a

* Trans. I.N.A., Vol. LIII. (Part II.), page 219.

resulting cavity, is equal to or greater than the velocity actually required for the influx of the water to the propeller blade. If we increase the number of revolutions at a certain speed of advance, and thus increase the slip so that the possible velocity of influx due to the pressure of atmosphere, gases, and water is below that required, the propeller gets an insufficient supply of water, and the occurrence of cavitation is brought about.

The criterion of cavitation depends on the required rate of supply to the blade being in proportion to the possible velocity of this influx. The influx velocity to the propeller blade, as long as cavitation does not set in (and neglecting surface friction) is normal to the surface and $= A B = v \cos a$ (see Fig. 1).

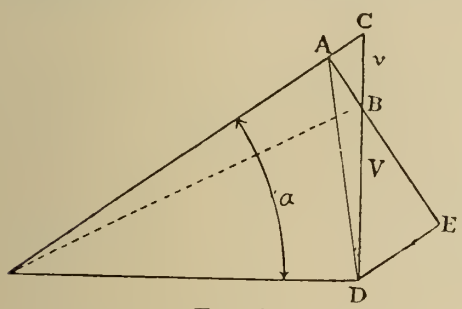


FIG. 1.

The maximum possible influx velocity $A B = v \cos a$ can be calculated under the condition that every particle of water passing through the propeller blade must be brought to the velocity A D resulting from that of the speed of advance $V = B D$ and that of the influx $A B = v \cos a$ by the speed V, the pressure of the atmosphere p_b above that which would exist in a cavity p_g due to water tension and gases, and by the pressure $D w$ of the column of water above the blade, where D = depth of water to propeller w = weight of water per unit of volume.

Hence, according to Bernouilli's theorem—

$$\frac{V^2}{2g} + \frac{p_b}{w} - \frac{p_g}{w} + D = \frac{(V \sin a)^2}{2g} + \frac{(V \cos a + v \cos a)^2}{2g}$$

$$v^2 + 2 V v = \frac{2g\left(\frac{p_b - p_g}{w} + D\right)}{\cos^2 a}$$

Table I. is calculated for a blade element with a pitch angle $a = 16° 20'$, and for the pressure—

$$\frac{p_b - p_g}{w} = 20 \text{ ft.},$$

and D = 6 ft. for various speeds of advance V, the slip limit being

$$\frac{v}{V + v}.$$

The most important result observable is that, *as the speed of the vessel increases, the limit of slip which can take place without cavitation is reduced. Further, that cavitation will start on the tip of the blade, where the pitch angle a is a minimum.*

For the purposes of the above inquiry, an infinitely thin propeller blade has been imagined. If we take a real blade into consideration, then we must introduce into our

formula the angle corresponding to the greatest pitch of the blade. This is generally the pitch angle at the emersion edge on the suction side of the blade. (Fig. 2.)

Let ε be the angle formed between the tangent of the following edge and the face of the blade in the cross section of the blade, the said pitch angle $= \alpha + \varepsilon$, then our criterion will be—

$$v^2 + 2\,\mathrm{V}\,v = \frac{2\,g\left(\frac{p_b - p_a}{w} + \mathrm{D}\right)}{\cos^2(\alpha + \varepsilon)}$$

The limit of allowable slip is thus arrived at—

$$s = 1 - \frac{\tan(\alpha + \varepsilon)\,\mathrm{V}}{\tan\alpha\,.\,(v + \mathrm{V})}.$$

If we suppose $\varepsilon = 8°$, and take the same conditions for granted as are stipulated in Table I., we then obtain Table II.

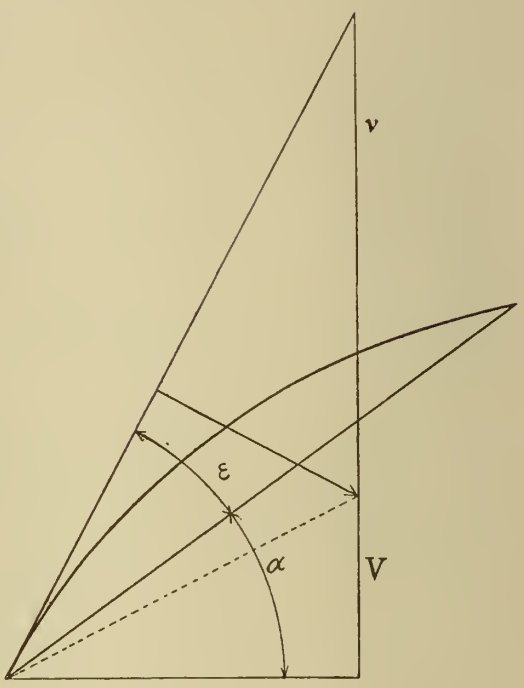


FIG. 2.

TABLE I.

Speed of advance V in knots.	V ft./sec.	V + v ft./sec.	Slip speed v ft./sec.	Limit slip.
0	0	42·65	42·65	1·000
4	6·76	43·18	36·42	0·845
8	13·52	44·72	31·00	0·693
12	20·28	47·20	26·92	0·570
16	27·04	50·48	23·44	0·465
20	33·80	54·40	20·60	0·379
24	40·56	58·80	18·24	0·310
28	47·32	63·80	16·48	0·258
32	54·08	68·80	14·72	0·214
36	60·84	74·30	13·46	0·181
40	67·60	79·85	12·25	0·154

From this it will be seen that, in order to prevent cavitation in the examples given, the propeller blade element must work with a negative slip at the rate of 24 knots, whereas in the case of the infinitely thin propeller, imagined for Table I., a positive slip of 31 per cent. would still have been admissible, at a speed of 24 knots. By a comparison of Tables I. and II., a *very clear idea can be formed of the enormous influence exercised by the form of the cross section of the propeller blade, and more especially by the angle* ε at the emersion edge. One realises the importance of making the feather-edge angle ε as small as possible, and, hence, of forming a cross section of the blade as nearly as possible approaching Fig. 3 (Plate XIII.).

TABLE II.

Speed of advance V in knots.	V ft./sec.	V + *v*. ft./sec.	$\frac{\text{Tan}\,(\alpha+\varepsilon)}{\text{Tan}\,\alpha}$	Limit slip.
0	0	44·90	1·543	1·000
4	6·76	45·45.	—	0·770
8	13·52	46·90	—	0·555
12	20·28	49·30	—	0·365
16	27·04	52·40	—	0·202
20	33·80	56·20	—	0·072
24	40·56	60·50	—	− 0·038
28	47·32	65·30	—	− 0·120
32	54·08	70·30	—	− 0·188
36	60·84	75·60	—	− 0·242
40	67·60	81·10	—	− 0·286

A practical test of the results arrived at can only be made by actual experiment. Many observations have, indeed, been made of the phenomenon of cavitation, but only a few measurements are available. The only measurements known to me are those of Mr. D. W. Taylor, who tested a screw propeller with a diameter of 16 in. and a pitch of 6·4 in. at three speeds of 5, 6, and 7 knots respectively, under various conditions of slip, from − 0·3 to + 0·4. The pitch angle at the tip of this propeller blade was $\alpha = 7^\circ\ 20'$.

The angle ε between the tangent at the following edge and the face of the blade is $= 21^\circ\ 50'$; if, as above, we take

$$\frac{p_b - p_g}{w} = 20 \text{ ft.}$$

and neglect D, we arrive at the criterion for the setting in of cavitation—

$$v^2 + 2\,V\,v = \frac{2\,g\,20}{\cos^2 29^\circ\ 10'}$$

Hence, we get at—

5 knots	a limit slip of			11·1	per cent.
6	..	..	..	− 2·6	..
7	..	..	..	− 22·0	..

A comparison with Taylor's experiments (Fig. 4, Plate XIII.) shows that, as a matter of fact, *we can take the setting-in of cavitation for granted at the above speeds, with the slip values given.*

If, from the known values of the thrust T and the torsion moment Q we calculate the abstract values—

$$\frac{T\,g}{\frac{d^2\,\pi}{4}\,w\,p^2\,n^2} \quad \text{and} \quad \frac{Q\,2\,\pi\,g}{\frac{d^2\,\pi}{4}\,w\,p^3\,n^2},$$

and coordinate these with the corresponding slip values, then, as long as no cavitation occurs, and no matter what the speed of advance of the propeller may be, we obtain two lines which can be taken as characteristic of the thrust and torsion moment respectively. *These lines are the same for all geometrically similar screws, of any desired dimensions, and for any desired fluid, say, for water or air.* These characteristics for thrust and torsion moment run almost in straight lines, as experience shows, up to slips of about 40 per cent. and in such a way that the following relations hold good (compare Fig. 5) :—

$$\frac{T\,g}{\frac{d^2\,\pi}{4}\,w\,p^2\,n^2} = a + b\,s \quad \text{and} \quad \frac{Q\,2\,\pi\,g}{\frac{d^2\,\pi}{4}\,w\,p^3\,n^2} = c + d\,s,$$

from which we get—

$$\text{efficiency } e = \frac{(a + b\,s)\,(1 - s)}{(c + d\,s)}.$$

As soon, however, as cavitation sets in, these characteristic lines alter by taking a considerable drop.

In Fig. 6 (Plate XIII.), for instance, the thrust and torsion moment characteristics for a twin-shaft turbine vessel are given, from which it will be seen that cavitation sets in when a slip of about 15 per cent. is reached.

Taylor's experiments, illustrated in Fig. 4, give entirely similar results. It may be

observed that, so long as no cavitation sets in, the propeller thrust and torsion moment are determined by the linear equations —

$$\frac{T g}{\frac{d^2 \pi}{4} w p^2 n^2} = 0{\cdot}430 + 0{\cdot}874\, s$$

$$\frac{Q\, 2\pi g}{\frac{d^2 \pi}{4} w p^3 n^2} = 0{\cdot}750 + 0{\cdot}492\, s$$

and that the deviation of the lines downwards occurs, in reality, at the slip values worked out above :—

$s =$ 0·111 for V = 5 knots.
$s = -$ 0·026 for V = 6 knots.
$s = -$ 0·220 for V = 7 knots.

Hence Taylor's experiments absolutely confirm the criterion advanced above.

We may, therefore, conclude—

(1) That the setting in of cavitation in screw propellers is not dependent on the mean surface pressure on the blade area, and not on the tip speed of the propeller blades.

(2) The finer the pitch of the screw propeller and the greater the angle formed by the feather-edge, and the greater the speed of advance, the less the slip speed should be to prevent the formation of cavities.

(3) That cavitation sets in earlier in a rough sea, when air is mechanically mixed with the water.

A lesser angle at the emersion edge will be more easily obtained in broad-bladed propellers than in narrower ones. In this fact lies, in my opinion, the secret of the success that has been obtained with turbine screws with broad blades. Since, however, broad blades have a lower efficiency than narrower ones of the same proportional section, in consequence of the increased friction, an attempt should be made, by distributing the cross section in an appropriate manner as in Fig. 3 (Plate XIII.), to construct as narrow propeller blades as possible by consideration of strength, with as small a feather angle as is allowable at the emersion edge. *Broad propeller blades should especially be avoided where, as in the case of merchantmen and battleships, the vessel's speed is kept below the limit at which, under ordinary circumstances, cavitation is not to be anticipated.*

The criterion here put forward has of course nothing to do with the cavities caused by the narrowing of the through-passage section between two blades, the presence of which is often shown by the occurrence of erosion especially near the root on the thrust side of the blades. The occurrence of these cavities is worthy of special study.

Written Contributions to the Discussion.

Mr. D. W. Taylor, Naval Constructor, U.S.N. (Member): Dr. Gümbel's theory of the cause and cure for cavitation is novel in that he considers primarily the following edge of the propeller blade, while practically all other theories that I am acquainted with deal mainly with the leading edge, or the whole body of the blade.

Experimental investigations made by me, and published at the same time as those used by Dr. Gümbel, appear to me to indicate conclusively that for the propeller blade of ordinary section, cavitation would begin on the driving face just back of the leading edge, just as soon as on the back of the blade just forward of the leading edge; also that the harmful effects of cavitation are due almost entirely to the cavitation on the driving face.

My experimental investigations also led me to the conclusion that for a given blade the primary factor of cavitation was its speed through the water. In Fig. 4 of Dr. Gümbel's paper (Plate XIII.), it is not not very easy to identify the exact point in each case at which cavitation may be said to begin, but if the curves were replotted, not on slip but on the actual tip speed, in each case, it would not be found difficult to maintain that the beginning of cavitation occurred at a given tip speed irrespective of the speed of advance.

I am inclined to think that the blade section proposed by Dr. Gümbel, and illustrated in his Fig. 3, would be a failure as regards the prevention of cavitation. The blade of a propeller is, after all, a submerged solid advancing through water, and the stream-line actions around it are materially different from what they would be in the case of an ideal blade of no thickness.

With the type of blade proposed by Dr. Gümbel, where the greatest thickness is forward of the centre, and both face and back of the blade are very convex in the vicinity of the leading edge, I think that at a comparatively low tip speed the water would break away from the blade on both face and back, somewhere near the point of maximum thickness, with the result that all the following portion of the blade would be working in a cavity, and the details of its shape would be wholly immaterial.

Mr. G. S. Baker (Member): Professor Gümbel's paper is one which is full of interest to those who are concerned with the propulsion of ships. This question of cavitation is becoming more prominent every day—not only on such vessels as destroyers, but on our large liners and other high speed vessels, and the thanks of the Institution are due to Professor Gümbel for bringing this paper before them. It represents a great deal of thought and patient investigation of experimental data, and requires careful reading and re-reading to grasp it. The subject interests us very greatly at the Froude Tank, and I venture to send the following notes for Professor Gümbel's consideration :—

The difficulty in applying theoretical investigations to experimental data is the proper evaluation of the constants involved, and the author has had to assume a value for $\frac{(p_b - p_v)}{w}$ on page 120. The figure taken is a very probable one, and for the purpose to which it is first applied serves as well as any other, but this is not necessarily the case when absolute quantitative results are required. Recent experiments at the National Physical Laboratory* show very conclusively that the pressure in the suction area about an air blade depends almost entirely on the tip velocity, and

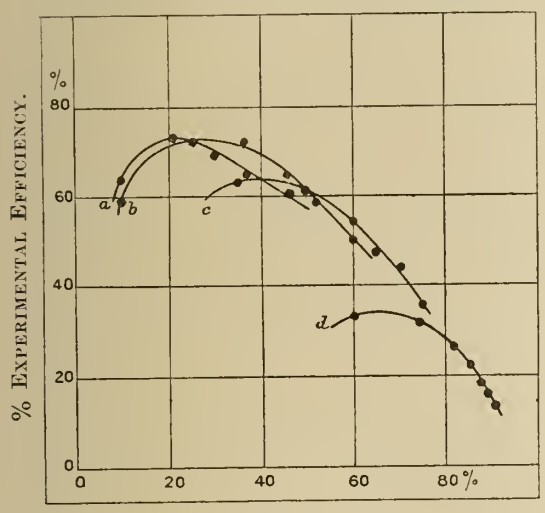
* Report of Advisory Committee for Aeronautics, 1911-12.

this term $\frac{(p_b - p_g)}{w}$ is a function of velocity. The experiments by Herr G. Fuhrmann* with a propeller in air give considerable support to the application of these blade results to actual propellers.

There are two critical slips with any propeller. These depend upon the pitch of the driving surface, the shape of the blade section, and the velocity of translation. At one of these critical slips the pressure per unit area on the face tends to a fixed value, and at the other the suction on the back breaks down and tends also to a fixed value,† these fixed values depending on the tip velocity.

Professor Gümbel's investigation applies only to this latter breakdown, but cavitation does not necessarily occur at this point. It will be present under the assumptions made by the author, but in an actual screw the suction pressure realised at this breakdown may not reach that necessary for cavitation. This is what occurs with air propellers which have the same characteristics in their efficiency curves, but as the suction does not, as a rule, exceed more than two or three feet water pressure, no cavitation occurs. But the variation in efficiency with velocity which is shown by the thrust curve on Plate XIII. does occur at moderate slips, as will be seen from the diagram (Fig. A), and in this case it is *not* due to cavitation. For these reasons I do not think the author has found the complete solution of cavitation, and it would be dangerous to adopt the section of blade given in Fig. 3 (which is not an efficient type) without reliable experimental data as to its action. Although differing in the above points from Professor Gümbel, I can admire his work, and the spirit of inquiry which has prompted it. We have learnt much from our German friends, and always welcome such a paper as this one.

Clark Air-Propeller Model B.
National Physical Laboratory Tests.



% Slip.

a. Translational Speed 2,800 f.m.
b. ,, ,, 2,000 f.m.
c. ,, ,, 1,200 f.m.
d. ,, ,, 400 f.m.

Fig. A.

Mr. S. J. Reed : I should like to add a few remarks to Dr. Gümbel's interesting paper. With regard to the section proposed for the propeller blades, there seems to be no doubt that from a hydrodynamic point of view this is very superior to the usual section, although it appears to me that the true helix should be through the centre of the section, and this seems to conform with Dr. Gümbel's first condition on page 118 for the occurrence of cavitation.

A practical disadvantage of this type of section is that neither half of the mould can be swept up, and both sides would have to be built up instead of only one, as with the section usually adopted. There would be less building up to be done, however, with the section given by Dr. Gümbel than if the true helix were through the centre.

Another practical disadvantage of this section is that the rough cast propeller could not be so easily trued up to pitch. However carefully the moulders do their work, propellers as they leave the mould

* Zeitschr. für Flugtechnik und Motorluftschifffahrt, April 26, 1913.
† *Liverpool Journal of Commerce*, May 8, 1913.

are never dead true to pitch, as, apart from any errors in moulding, they very often alter in cooling. It is now the almost invariable custom, with high-speed propellers, to either machine or chip the face of the blades true to pitch, as it is generally recognised that with high speeds a very small variation in pitch of the blades causes vibration. A case of vibration due to the pitches of the blades of the propeller not being quite accurate has come within my own experience, and the extent of the variation in pitch of this particular propeller was only very small.

Dr. Gümbel, in summing up his conclusions, is in disagreement with Mr. Barnaby's view that cavitation is dependent on the pressure over unit area of the blade. However, it appears to me that Mr. Barnaby's limitation of the thrust on the blades is a practical way of guarding against an excess of slip as given in the author's third condition on page 118. The limitation of the slip as a function of the ship's speed certainly seems to me to be the primary cause of cavitation, and after a certain point is reached it is impossible to prevent it occurring, owing, as Dr. Gümbel stated, to there being a limiting velocity at which the water flows in to the propeller to take the place of that discharged astern.

The section of the blade and the speed through the water no doubt do affect cavitation. It is, however, purely a secondary cause, and if it were possible to make the propeller blades so that they had no thickness, and also to eliminate the skin friction of the water, it would disappear altogether.

Although in practice, owing to the disturbed water in which the propeller works, and also due to the effect of the following wake, propeller design is very involved, it seems that exactly the same principles apply to a propeller as to an ordinary pump. The only difference is that in the case of the pump the axial thrust produced on the impeller due to pumping water is of no service, and must either be balanced hydraulically or else the thrust must be taken on a thrust bearing. With the propeller, however, the only useful action is the axial thrust, and the actual pumping of the water astern is only carried out in order that this axial thrust may be produced.

The axial thrust on a pump impeller or a propeller is mainly due to the difference in pressure of the water at the two sides acting over the area exposed. Now at the suction side there is a limit to which the pressure can be reduced, depending on the temperature of the water. For instance, if the temperature is 50°, the limit to the reduction in pressure is 0·18 lb. per sq. in. absolute. At this pressure, with the temperature given, the vapour tension of the water is reached, and, as it cannot remain as such, it will immediately commence to boil. This means that, instead of the pump or propeller drawing water, it sucks in steam or vapour; that is to say, a vacuous space is formed.

The limit to the reduction in pressure with a given temperature is generally made evident to engineers very early in their career, in connection with feed pumps, as, if for any reason the temperature of the hot-well becomes too high, immediately the water reaches the low pressure at the suction side of the pump cylinder it is converted into steam and the pumps start hammering.

If the absolute static pressure of the water, due to the head and atmospheric pressure is known, the theoretical velocity at which the water will run into the reduced pressure space in front of the propeller to take the place of that discharged astern can easily be calculated; as the two are dependent on each other, however, the calculated velocity would have to be considerably modified for friction, &c.

When dealing with the propeller, from the point of view of differences in pressure, Mr. Barnaby's limitation to the thrust over the unit area of the blade appears to be quite a rational method of guarding against the primary cause of cavitation.

There is no doubt that with similar revolutions and pitch an increase in the surface enables a greater thrust to be developed by the propeller. This was shown in a marked manner in some experiments which I carried out some years back. These experiments were only on a very small scale, as experiments go, but the results obtained appeared to agree fairly well in practice.

Tip speed and blade section in themselves appear to be no guide for the occurrence of cavitation ; for instance, four different propellers which have all come within my own experience, although different in other dimensions, all had a pitch ratio of approximately ·93. All these propellers appeared to show signs of cavitation at top speeds, however, the tip speeds varying as widely as 140 ft. per second and 200 ft. per second. Certainly the apparent pressure per square inch of surface varied a good deal also, namely, from 12½ lbs. to 15 lbs. The lowest pressure was with a propeller having, when compared with the others, rather narrow thick blades, which seems to point to the fact that blade section was largely responsible for a higher pressure not being obtained.

The calculation of the thrust developed is a very uncertain quantity, however, and for this reason I think that it would be of very great benefit if this thrust were actually measured on trial trips. This would not only enable valuable and reliable information to be obtained in connection with propellers, but would also serve to check the calculations for the thrust required to propel the ship.

I do not know whether the system put before the Institution by Mr. Heck in 1909* has ever been applied. I once knew of preparations being made for it, but it was not carried out. I think that some hydraulic system such as Mr. Heck's is the one that is the most likely to give accurate results. The only time I have known an attempt to be made to obtain it by measuring the compressive strain on the shaft, it proved a failure.

There is just one other point I should like to mention in order to show that cavitation has more to do with what I have called the "primary cause," namely, the limit to the pressure to which the water can be reduced on the suction side, thus limiting the velocity of inflow, rather than to blade speed or section. As everyone who has had to deal with turbine ships knows, owing to these being fitted with small diameter propellers which are very nearly at cavitation point under full-power conditions, they are not so easily handled as those in which the diameter of the propellers is comparatively large and with which the cavitation point is not nearly reached under full-power conditions.

With the latter class of propeller it is possible to develop a far greater thrust when the ship is stationary than when it is under way ; but with the high-speed propeller fitted to directly coupled turbine ships, owing to the appearance of cavitation, the limit to the thrust that the propeller can produce is very quickly reached. Owing to this, there appears to be very little advantage, for the sake of the manœuvring qualities of the ship, in increasing the power of astern turbines beyond a certain percentage of the ahead ones, as the maximum thrust can be produced with the expenditure of far less power when the ship is stationary or moving slowly, than when she is under full-power conditions. This is due to there being no velocity of inflow of the water to the propeller when the ship is stationary, such as is created by the motion of the ship when she is under way. On more than one occasion I have tried to screw a ship round more quickly by developing as much power in the engine-room as possible, but have found that after a certain number of revolutions were obtained no further advantage was gained.

Although the adoption of transmission removes the very acute stage with propellers, the question still remains a very pressing one as to whether the principle employed shall be mechanical, hydraulic, or electrical. From the consideration of weight, cost, &c., it is invariably desirable to run the propeller at as high a speed as possible in order to keep the transmission ratio as small as possible. Even when considering transmission, Dr. Gümbel's paper is therefore very welcome in assisting to put propeller design on a rather more satisfactory basis than it is at present.

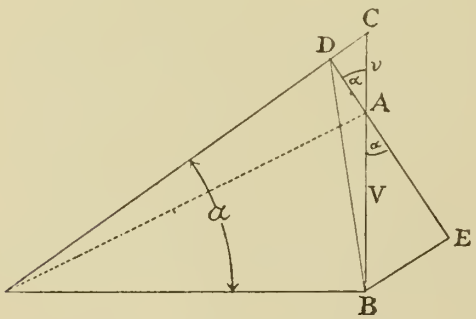
Mr. E. R. Mumford (Member) : Dr. Gümbel commences his paper by a particular reference to Mr.

* Trans. I.N.A., Vol. LI. page 87.

Sydney W. Barnaby's criterion of 11 lbs. per square inch of projected blade area against which he quotes the published results of actual observations by Mr. D. W. Taylor, and then proceeds to deal with the matter by his own method, which entirely ignores pressure per unit area as a criterion for the occurrence of cavitation.

Dr. Gümbel is probably aware that Mr. Barnaby describes "Cavitation" as a phenomenon, the manifestation of which depends on the speed at which water can follow up the blades of a screw. There is probably no difference of opinion on that point. Mr. Barnaby has, however, published his conclusion, based on the results of the trials of H.M.S. *Daring*, that a thrust of 11¼ lbs. per square inch of projected area of propeller blade is "therefore about the maximum thrust which can be obtained from a screw working efficiently at a depth of 11 in. below the surface." Mr. Barnaby also said that the figure may vary slightly with the variation of pitch ratio and shape of blade. Also, that for every additional foot of immersion the total thrust per square inch may be increased by three-eighths of a pound. This criterion of pressure per square inch was clearly intended by Mr. Barnaby to provide a convenient measure for limiting the speed of flow of water to that which should not be exceeded in order to avoid undue cavitation, and it is this criterion of thrust per square inch of blade area that is not accepted by Dr. Gümbel, in common with Mr. D. W. Taylor, as quoted by Dr. Gümbel, and Messrs. Wm. Denny & Bros.* Mr. Barnaby has subsequently raised the permissible thrust per square inch of projected surface to 13 lbs. per square inch for turbine-driven vessels, giving as his reason for doing so, that the turning effort of turbines is more uniform than that of reciprocating engines, and that a certain amount of cavitation may be permitted.

In the discussion before this Institution already referred to, Sir Archibald Denny quoted much higher pressures per projected area as being associated with good propulsive efficiencies; while Mr. D. W. Taylor, as quoted by Dr. Gümbel, refers to instances of cavitation with a thrust of only 4 to 5 lbs. per square inch. These variations show that thrust per unit area is not the only important factor that determines the rate of flow of water towards the propeller, and, therefore, should not be relied on as a safe criterion of cavitating conditions.



Mr. Taylor attaches importance to the speed of the propeller blade through the water, and to the shape of the blade, as controlling factors, while Dr. Gümbel selects pitch of cross section, and particularly the pitch of the back of the blade at the following edge; also the amount of the slip as a function of the vessel's speed, temperature, degree of air saturation, barometric pressure, and immersion, as factors which affect to a varying degree the rate of flow towards a propeller.

Probably all those who have seriously considered the subject will agree that area, shape, and cross section of blade, absolute speed of ship and of blade along its path through the water, temperature, and gas absorbed in the water, barometric pressure and immersion, are all factors which more or less affect the possible rate of flow towards a propeller for a given thrust per unit disc area; the difficulty is to determine the value of each factor. Dr. Gümbel, however, introduces a new consideration, and claims to prove that "as the speed of the vessel increases the limit of slip which can take place without cavitation is reduced." His proof, however, seems to depend upon the first equation on page 119, where he appears to have

* See Dr. (now Sir Archibald) Denny's remarks in the discussion of Mr. Barnaby's paper, Trans. I.N.A., Vol. LIII. (Part II.), pp. 226 and 227.

fallen into an error by equating the sum of two pressure heads, corresponding to two separate speeds, to the pressure head corresponding to the sum of the two speeds.

This equation can be dealt with more satisfactorily by slightly adding to and lettering Fig. 1 of Dr. Gümbel's paper, as shown on the opposite page.

D B = the resultant speed of flow of water from the propeller.

$$(\mathrm{D\ B})^2 = (\mathrm{B\ E})^2 + (\mathrm{A\ E} + \mathrm{A\ D})^2 = (\mathrm{V} \sin \alpha)^2 + (\mathrm{V} \cos \alpha + v \cos \alpha)^2.$$

Therefore the head of water which will give a speed of D B is equal to

$$\frac{(\mathrm{V} \sin \alpha)^2 + (\mathrm{V} \cos \alpha + v \cos \alpha)^2}{2g}$$

and this is the quantity on the right-hand side of Dr. Gümbel's equation.

On the left-hand side of the equation Dr. Gümbel has added $\frac{\mathrm{V}^2}{2g}$, which is the head required for a speed V, to the head

$$\left(\frac{p_b - p_g}{w} + \mathrm{D}\right),$$

which represents the effective head of atmosphere and water, and is the pressure head available for compelling the water to keep touch with the propeller at the speed of the slip. Thus the sum of the two pressure heads, that give the speed V and (D B − V) respectively, is equated to the pressure head that gives the total speed D B, and this is incorrect, because pressure head varies as the square of the speed, so that while the sum of the two pressure heads is equal to

$$\frac{\mathrm{V}^2 + (\mathrm{D\ B} - \mathrm{V})^2}{2g}$$

the pressure head required for the combined speed is equal to

$$\frac{\{\mathrm{V} + (\mathrm{D\ B} - \mathrm{V})\}^2}{2g} = \frac{(\mathrm{D\ B})^2}{2g}$$

Thus the left-hand side of Dr. Gümbel's equation should be corrected to

$$\frac{\left\{\mathrm{V} + \sqrt{2g\left(\frac{p_b - p_g}{w} + \mathrm{D}\right)}\right\}^2}{2g}$$

which represents the square of the sum of the speeds due to the two pressure heads divided by $2g$.

The equation thus corrected does not appear to permit the deduction that "as the speed of the vessel increases the limit of slip which can take place without cavitation is reduced," for it can be resolved into

$$2\,\mathrm{V}\left\{\sqrt{2g\left(\frac{p_b - p_g}{w} + \mathrm{D}\right)} - v \cos^2 \alpha\right\} = v^2 \cos^2 \alpha - 2g\left(\frac{p_b - p_g}{w} + \mathrm{D}\right)$$

By substituting proper values for

$$v \text{ and } \frac{p_b - p_g}{w} + \mathrm{D},$$

whieh are definitely related for eaeh angle—the relation varying with angle beeause the differenee between (D B – V) and v varies with variation of angle—it will be found that the equation beeomes $2\ V \times O = 0$.

The method of assuming a non-advaneing propeller with water flowing towards it at a speed equal to the speed of the ship, and then finding the pressure head eorresponding to that speed, &e., is not one that ean be eommended for its simplieity, as it not only is liable to lead to eonfusion and error, but it also involves considerations of pressure heads whieh do not exist. It is relevant to point out that the only pressure head available is that of the atmosphere and water above the propeller, and that the pressure is only suffieient to aeeelerate the water mass to the speed of the slip when eavitation is commeneing—thus the speed V of the ship is not impressed upon the water, and requires no pressure head, so that $V = 0$ when dealing with speed impressed on the water, thus Dr. Gümbel's equation really beeomes

$$\frac{p_b - p_v}{w} + D = \frac{(v \cos \alpha)^2}{2g}$$

whieh, while correet, is not helpful in establishing a eriterion for the oeeurrence of eavitation.

The faetors whieh affeet the mass of water acted upon by the propeller, and therefore the speed of the slip for a given thrust, are usually assumed to be diameter, blade area, and speed of ship. An inerease of either of these faetors will give a less speed of slip to the larger mass for the same thrust. There is, however, another important faetor whieh affeets the mass of water aeted upon, though it appears to be generally omitted from eonsideration, and that is, revolutions per distanee, or revolutions per minute for a given speed of ship. The greater the number of revolutions per minute for a given speed of ship, the more eompletely does the propeller aet upon the water covered by the dise; thus an infinite number of revolutions per minute would make the blade area equivalent to that of a complete dise, and eavitation would take plaee at a relatively high pressure per unit blade area, while with no revolutions per minute (*i.e.*, assuming a non-revolving propeller to be pushed aft from the stern of the ship at a speed of V + v) the propeller eould not aet upon a greater proportion of the disc than is covered by the projected area of the blades; only in sueh a case as this would pressure per unit of projected blade area be a eorrect eriterion for speed of slip, and the oeeurrence of eavitation; and sueh a ease would give the minimum pressure per unit area at whieh cavitation eould oeeur. For very high revolutions per minute, pressure per unit dise area is a much better eriterion than pressure per unit blade area. The differenees between the revolutions per minute of various vessels of about the same speed are the prineipal eauses of the differences in cavitating pressures per unit blade area; thus H.M.S. *Daring* was a 24-knot ship, with propellers driven by reeiproeating engines at about 380 revolutions per minute, and they broke down at a pressure of 11¼ lbs. per square ineh of projeeted area. Vessels of about the same size and speed, with propellers driven direet by turbines (*i.e.*, no redueing gear), would have about double the revolutions per minute of the *Daring's* propellers, and would, therefore, be able to aet more effectively on the water eovered by the disc, and so have less slip for the same thrust per unit area of blade, and earry a greater thrust per unit blade area for a similar degree of eavitation.

The turning moment of the reciprocating engines of the *Daring* doubtless varied eonsiderably during eaeh revolution, and a proportion of the variation was probably eonveyed to the propellers, notwithstanding the spring of the shafting. Sueh variation of turning moment would obviously cause a breakdown due to eavitation at a less pressure per unit blade area than in the case of propellers driven by the uniform turning moment of turbine engines.

There are other eontributory eauses of the variations in pressure per unit blade area for the same degree of cavitation at the same speed of ship, namely, variation in radial disposition of blade area,

variation in shape and thickness of blade section, variation in temperature and the amount of gas absorbed in the water, variation in barometric pressure, and immersion.

When the speed of ship varies the amount of water acted upon by the propellers per unit time varies—the higher the speed the greater the mass, and the less the slip for the same thrust; or the greater the thrust per unit blade area that can be carried for the same degree of cavitation.

Numerous results of actual trials of ships, on the Skelmorlie measured mile, show that in those waters the phenomenon of cavitation, in the case of propellers of fine pitch ratio, will generally be manifested when the speed of real slip* is about 16 ft. per second, though theoretically the pressure head of atmosphere and water should permit of about three times that slip against a zero pressure, the factors of gas absorption, temperature, &c., probably contribute largely to the great difference between the theoretical and the actual, the ebullition of gases and water vapour in the cavity creating a considerable pressure therein. This limit speed of slip, however, is much higher than Dr. Gümbel has worked out in Table II. of his paper, and in very many cases it permits of a much higher pressure per unit of projected blade area than Mr. Barnaby has conceded, due to the causes referred to, not the least of which is the relatively high revolutions per distance.

The most reliable criterion for the occurrence of cavitation is actual experiment with model propellers working behind the ship model fitted with shaft brackets, at a speed of advance and thrust corresponding to ship requirements. Such experiments will integrate the varying effects of the principal factors—diameter, blade area and shape, pitch, and revolutions per minute—so that, although the model conditions are far removed from cavitating conditions, yet when model results show that a real slip* of more than 16 ft. per second would be required for the ship's propellers, it is at once a sign of danger which requires to be met by increasing diameter, or blade area, until the model results show that the corresponding speed of the real slip* of the ship's propellers will be less than 16 ft. per second, and that, therefore, undue cavitation will be avoided. It often happens, however, that in preparing a design with propellers having the minimum pitch ratio permissible, somewhat higher speeds of real slip than 16 ft. per second may be deliberately incurred, for the purpose of increasing the revolutions per minute of the turbines, thus permitting a certain degree of cavitation with consequent reduction of propulsive efficiency. In all such cases the degree of cavitation must be so limited that the loss in propulsive efficiency is less than the gain in turbine efficiency, or that due to reduction of weight.

The importance of the subject (particularly in these days of high speed, turbine-driven propellers) must be my apology for discussing this matter at such considerable length.

Sir Philip Watts, K.C.B., F.R.S., D.Sc. (Vice-President): Professor Gümbel's analysis of the phenomenon of cavitation expresses simply the characteristics of the motion, and the criterion formula he gives will be a valuable aid to those who are dealing with this troublesome quantity.

It is true that the thrust pressure is not, strictly speaking, a true measure of the cavitating speed: but it is a measure which can be readily used for comparing similar screws under similar conditions, and if I remember rightly, it was under such assumptions that Mr. Barnaby first suggested its use.

If, however, the question be considered generally, the conditions enumerated by Dr. Gümbel are

* Real slip, in feet per second, as distinct from apparent slip, is equal to $(P \times R) - (S - W)$, where P = pitch, in feet, of driving face of propeller blades, R = Revolutions per second, S = Speed of ship in feet per second, and W = Forward speed of following wake in feet per second. It should be remembered, however, that the true pitch of a propeller blade usually differs from the pitch of the driving face; in the case of a blade of fine pitch ratio the true pitch is considerably greater than the pitch of the driving face, and therefore the actual cavitating speed of slip is correspondingly greater then that deduced by assuming the pitch of the propeller to be that of the driving face.

doubtless correct. The cavitating limit is governed by absolute velocity of blade, slip, and cross-section of blade. It is obvious that his condition No. 4, page 118, must apply to any method of investigation. The dominant factors are, I think, Nos. (1), (2) and (3).

The comparison between a thin blade, Table I., and a practicable blade, Table II., is made by taking only the face pitch angle and the extreme pitch angle of the back. The differences in the Tables show very clearly the enormous effect which actual blade shape and thickness have on cavitation. It appears to me, however, that to take the maximum pitch angle of the back is to go to an extreme, and that probably the truth lies somewhere between the two. Dr. Gümbel's analysis, moreover, does not take account of one possibility, viz., that between the thin plane and the actual blade the distribution of velocity is probably very different, and that cavitation may be set up more by this velocity or pressure distribution than by an alteration in pitch angle.

The proposed section given in Fig. 3 (Plate XIII.), viewed in the light of this paper, is certainly likely to raise the limit for cavitation; but before adopting it, it would be necessary, I think, to make thorough experimental investigations. Actual speed of propeller blade is a good guide to cavitation, but beyond this, the question of the shape of the blade section appears to me to be less simple than the author appears to have considered it.

The thanks of the Institution are due to Dr. Gümbel for his paper, which, I am sure, will be of much interest to our members.

Dr. L. Gümbel (Member): Allow me to answer in their proper order the various observations which have been made on my paper on "The Occurrence of Cavitation."

Mr. D. W. Taylor considers that it is difficult to determine from Fig. 4 the point when cavitation actually sets in. In accordance with Mr. Taylor's suggestion, therefore, I have altered Fig. 4 so that the tip speed is utilised as axis of the abscissæ. In this manner we obtain a diagram which, it is true, shows the setting in of cavitation much more clearly than the original one of Mr. Taylor's (Fig. 220 in his work), but in my opinion not more so than does Fig. 4 in my contribution on the subject. The full value of the representation selected by myself is best realised when the torque arrived at by my method is compared with that resulting from Mr. Taylor's representation. I have added the three methods of representation for the purposes of comparison (Figs. 7 and 8, Plate XIV.). The representation of the torque is of much greater importance than that of the thrust, inasmuch as we are able, on board ship, to measure the torque without any great difficulty, by means of the torsion meter, whereas the determination of the thrust cannot generally be carried out. Fig. 6 (Plate XIII.) gives such measurements for a turbine-driven torpedo boat. It will be seen that the curve of the characteristic of the torque

$$\frac{Q\, 2\, \pi\, g}{\frac{d^2\, \pi}{4}\, w\, p^3\, n^2}$$

is entirely sufficient (see also Fig. 8) for the purpose of establishing the point at which cavitation sets in. This method of representation also enables one to ascertain the influence of cavitation on efficiency in the simplest possible manner. By that method it will be found that efficiency is influenced but to a relatively small extent by cavitation, inasmuch as with a slip of 25 per cent. the efficiency only drops to 47·8 per cent. from 52·5 per cent.

Mr. Taylor further states that "for a given blade the primary factor of cavitation is in the speed of the blade through the water," and that "the beginning of cavitation occurs at a given tip speed, irrespective of the speed of advance." In reality, Mr. Taylor's representation of the thrust values shows

that cavitation sets in at approximately the same tip speed. Dependence on the speed of advance only exists to a limited extent, in so far as, in the case of a lower speed of advance, cavitation sets in at a somewhat lower tip speed than it does when the speed of advance is greater. This result coincides completely with the criterion for the occurrence of cavitation given by myself, which is :—

$$v^2 + 2\,V\,v = \frac{2\,g\left(\frac{p_b - p_g}{w} + D\right)}{\cos^2(\alpha + \varepsilon)}$$

This expression can be simplified as follows :—

$$(v + V)^2 = \frac{2\,g\left(\frac{p_b - p_g}{w} + D\right)}{\cos^2(\alpha + \varepsilon)} + V^2$$

Since $v + V$ is proportional to the tip speed and independent of the speed of advance, we can also express the criterion for the occurrence of cavitation :—Tip speed proportional to

$$\sqrt{\frac{2\,g\left(\frac{p_b - p_g}{w} + D\right)}{\cos^2(\alpha + \varepsilon)} + V^2}$$

If we insert the numerical values obtained by Mr. Taylor in the course of his trials, we arrive at the following three equations for the three speeds of 7, 6, and 5 knots respectively :—

Tip speed proportional to $\sqrt{515 + 139{\cdot}2}$ or proportional to 25·58
$\sqrt{515 + 102{\cdot}4}$ or proportional to 24·85
$\sqrt{515 + 71{\cdot}2}$ or proportional to 24·24

which agree well with the result of Fig. 7 (Plate XIV.).

From this formula it will be seen that so long as the term V^2 is small in relation to the term

$$\frac{2\,g\left(\frac{p_b - p_g}{w} + D\right)}{\cos^2(\alpha + \varepsilon)}$$

the criterion of cavitation is practically independent of the speed of advance. For higher speeds of advance the proportionality disappears more and more. I think, also, that Mr. Taylor will surely agree with me that it would be a mistake to state that the occurrence of cavitation in all propellers is uniformly connected with a determined tip speed ; for, whereas in the trials now before us, carried out by Mr. Taylor, cavitation sets in at a tip speed of some 5,000 ft. per minute, it is well known that in the case of standard torpedo-boat propellers tip speeds of 15,000 ft. per minute are admissible. Mr. Taylor's trials, therefore, even if we consider them as dependent on the tip speed, rather tend to confirm than to refute the truth of the criterion advanced by me.

With reference to the further remarks of Mr. Taylor and Mr. G. S. Baker *re* the influx and the distribution of the pressure of the water on the propeller blade, I should like to point out the following : The propeller employed by Mr. Taylor in his trials, according to Fig. 221 of his work (reproduced in Fig. 9, Plate XIII.) is one that should never be used for driving high-speed vessels. If we make a drawing of this propeller section showing the course of flow to the propeller blade, we obtain results about the same as are shown in the diagram (Fig. 9). On the back or suction side of the blade, at the immersion edge C, there is pressure ; in other words, a force is called into play which is directly opposed to the effect we wish

to obtain. On the front surface of the propeller blade, both at A and at B, we get suction, which again is an effect which is diametrically opposed to the one we have in view. The only effective part of the propeller blade is the emersion edge on the suction side of the blade D, since only at that part does the force acting on the water coincide with the direction we wish it to take. Entirely similar effects occur, although not in the same degree, in the case of every blade in which the pressure side is constructed throughout its whole breadth as a true screw surface, when the slip is correspondingly small, as has been shown by the trials in the William Froude tank mentioned by Mr. Baker and by the experiments of the Göttinger Versuchsanstalt.*

The proper direction can only be given to the various forces by adopting the shape I have proposed, and which I give in Figs. 3 and 10 (Plates XIII. and XIV.). As will best be seen by reference to the diagram of the course of influx (Fig. 10), in the shape proposed the pressure which we had before on the immersion edge of the suction side of the blade has now gone over to the front side of the blade towards A. On the rear side of the blade we have suction effect at C and D. All the forces are, therefore, working in the desired direction. In my opinion it is not necessary to give increased sharpness to the immersion edge of the blade, as proposed by Mr. Taylor, inasmuch as the direction of the influx lines to the immersion edge on the suction side must in any case be diverted by the propeller blade, and sharpening the edge of the same could in no way alter these conditions, which are dependent solely on the development of the influx lines on the front side of the propeller blade. To form an idea of the influence exercised by the emersion edge on the thrust, the following considerations will be of service. If the emersion edge of the blade is obtuse (Fig. 11), vortices are formed at the emersion edge which draw off the water flowing along the front surface of the blade right away from the propeller, thus bringing about suction on the emersion edge of the front side of the blade at B, as can be seen by Fig. 11. The sharper the angle given to the emersion edge of the propeller blade, the less will be the formation of vortices and, therefore, the less suction we shall have on the front surface of the propeller blade; in other words, the less forces we shall have working against us.

To sum up, I hold that the pressure observed by Mr. Taylor on the immersion edge on the suction surface of the blade and the suction on the front surface are the results of faulty construction of the blade, but they are not unavoidable occurrences. It must ever be the constant endeavour of constructors to arrive at a propeller blade section which will yield the greatest possible suction throughout the back surface and, at the same time, the greatest possible pressure on its front surface, and I am of the opinion that a section such as is shown in Figs. 3 and 10 will fulfil these conditions in the best possible way.

To go into all the questions connected with the subject would involve too much detail and lead me far beyond the scope of my present contribution to the subject in hand. I would, however, like to refer to my coming paper on the subject before the "Schiffbautechnischen Gesellschaft" (November, 1913), in which I propose going exhaustively into the details of this problem.

With respect to Mr. S. J. Reed's remarks, I would like to state that the front side of the propeller blade section which I have proposed is constructed as a true screw. It is quite possible to construct this propeller on normal lines, so that only one side of the blade need be built up. The measurement of the pitch can, moreover, be carried out along *a a* as in ordinary propeller blades (Fig. 10).

It is somewhat more difficult to answer the objections advanced by Mr. Mumford. He considers the equation I gave to be incorrect, and endeavours to prove that instead of the equation being

* Zeitschrift für Flugtechnik und Motorluftschifffahrt, Mitteilungen aus der Göttinger Versuchsanstalt von G. Fuhrmann.

$$v^2 + 2\,\mathrm{V}\,v = \frac{2\,g\left(\frac{p_b - p_g}{w} + \mathrm{D}\right)}{\cos^2(\alpha + \varepsilon)}$$

it should be stated as follows :—

$$v^2 = \frac{2\,g\left(\frac{p_b - p_g}{w} + \mathrm{D}\right)}{\cos^2 \alpha}$$

so that the setting in of cavitation would result independently of the speed of advance. A certain brevity in the manner in which I expressed my train of thought is doubtless accountable for Mr. Mumford's not having been able to follow my meaning. I will, therefore, attempt to put before you again my line of thought as clearly as possible. Firstly, in contradistinction to Mr. Mumford, I take it as a recognised fact in Mechanics that, in estimating the forces in a propeller, only the relative speeds between the propeller and the water need be taken into consideration. We can, therefore, imagine the propeller as revolving about a fixed point, whereas the water arriving from infinity with a velocity V flows to the propeller with a velocity of V + v. The action of the propeller consists in the fact that on the back surface of the blade a certain partial vacuum is formed, and it is just upon the occurrence of this partial vacuum that the increase of velocity from V to V + v is actually dependent. All that is required is to solve the hydrodynamic problem of determining the velocity of the water under the influence of the partial vacuum formed on the blade surface. If we imagine the water as flowing through a tube, then we obtain the power of the water of doing work per unit of weight of water flowing through the tube through the kinetic energy $\frac{\mathrm{V}^2}{2g}$ and the potential energy $\frac{p_b}{w} + \mathrm{D}$. On flowing into the propeller the sum of these two energies remains unaltered. The potential energy in the propeller is equal to $\frac{p_g}{w}$. Therefore we find the kinetic energy of the water on entering the propeller to be (considering only the axial velocity for simplicity)—

$$\frac{(\mathrm{V} + v)^2}{2g} = \frac{\mathrm{V}^2}{2g} + \frac{p_b}{w} + \mathrm{D} - \frac{p_g}{w}$$

which is in general agreement with the equation I gave above.

But Mr. Mumford, in the further course of the discussion, says :—" When the speed of a ship varies, the amount of water acted upon by the propeller varies, the higher the speed the greater the mass, and the less the slip for the same thrust." Now, what Mr. Mumford says there is exactly what my equation expresses, since the thrust which can be exercised from the back surface of the propeller blade is constant as soon as cavitation sets in at the surface in question. Therefore, as Mr. Mumford very rightly remarks, the slip must be smaller at the surface in question when the speed of advance is greater.

Further, Mr. Mumford's contention is not borne out by practical results. As he very rightly remarks, cavitation sets in in ships at a speed of some 25 to 30 knots, when the slip speed is about equal to 16 ft. per second, but according to Mr. Mumford's reckoning it should set in at a 45 ft. slip speed. According to our Table I., the slip speeds at which cavitation sets in are :—For 24 knots, 18·21 ft.; 28 knots, 16·48 ft. According to Table II., for 24 knots, 19·94 ft.; for 28 knots, 17·98 ft.; in other words, in general accord with experience, but in any case in better accord than the 45 ft. calculated by Mr. Mumford as against 16 ft.

In the course of the present paper I have simply dealt with the occurrence of cavitation; its extent depends on the shape given to the propeller blade. An obtuse blade will give indications of cavitation

even at low speeds (Fig. 11, D and B), without affecting the working of the propeller to any great extent. In the case of turbine-driven propellers, too, the setting in of cavitation is not attended with any serious reduction in the efficiency of the propeller, as has been shown above. On the other hand, the rapid reduction of the thrust and of the torque is indissolubly connected with cavitation. The maximum thrust which can be exercised on the back surface of the blade comes into play when the full suction power $p_b - p_a$ occurs at that surface.

Mr. Barnaby's criterion is so far justified that by means of it it is possible to set a limit at which the back surface of the blade reaches its limit of suction power. It is, however, not a criterion for the setting in of cavitation. Just as little is it a criterion of the maximum thrust of the propeller, since besides the suction force on the back surface of the blade, if the section of the latter has been properly selected, we also have to take into account the pressure on the front surface of the blade.

The other remarks made by Mr. G. S. Baker appear to me to be too concise, unfortunately, to allow of my understanding their full significance. I hope Mr. Baker will, by presenting a detailed and comprehensive account of his researches, give us an opportunity to study his views and his experience in this domain. I will only refer to one point, therefore. Mr. Baker, if I understand him correctly, believes that phenomena did occur with the Clark air propeller model, which produced effects similar to cavitation, though there was no real cavitation. I am unable to draw that conclusion from the experiments conducted at the National Physical Laboratory. For when we plot their experimental results, as published in the Report for Aeronautics (1911-12), in the manner which I have recommended (Fig. 12, Plate XIV.), the ordinate being the slip and the abscissa the term

$$\frac{T \cdot g}{d^2 \frac{\pi}{4} \cdot w p^2 n^2},$$

we obtain curves of a perfectly regular character, which do not at all display that character which is observed with cavitation (Figs. 6 and 8, Plates XIII. and XIV.). It is true that these curves should coincide with one single curve for all translational speeds. That they do not coincide is, in my opinion, due to the circumstance that the results were obtained with the aid of a whirling arm, and that propeller experiments with the whirling arm can never be regarded as exact. For this reason the whirling arm method of experiment is not employed on this side of the Channel.

If Mr. Baker's results were perfectly correct it would be impossible to utilise propeller models for the calculation of real propellers, and impossible also to deduce a rational propeller theory. I am of the opinion that, if the experiments with the Clark propeller (model B) were repeated in water, the unreliability of the results obtained with the whirling arm would be recognised.

With regard to the remarks of Sir Philip Watts, I am glad to see that he agrees in general with my deductions. The consideration of the largest pitch-angle at the back of the cross-section of a blade is necessary, since I merely desired to investigate the conditions prevailing at the *setting in* or *commencement* of cavitation, without wishing to determine how cavitation spreads, and how it becomes detrimental. I also agree with Sir Philip Watts, that further experiments will be required in order to determine the most favourable form of cross-section of a blade; but I think that the following two leading rules must stand as a basis for these experiments :—

(1) The leading edge on the pressure-side (A in Figs. 10 and 11, Plate XIV.) must be brought up high (*i.e.*, be strongly curved) in order to prevent the occurrence of any suction effect at this point.

(2) The trailing edge on the suction-side (D in Figs. 10 and 11) must be as fine as will be at all compatible with the strength of the blade.

ON THE TRIALS OF THREE FERRY STEAMERS PROPELLED BY GEARED TURBINES.

By J. Inglis, Esq., LL.D., Hon. Vice-President.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913; Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland, in the Chair.]

The following paper is offered in response to a suggestion from Dr. Archibald Denny (now Sir Archibald Denny, Bart.), during the discussion on Sir Charles Parsons' paper read at the Spring Meetings. Dr. Denny expressed a wish to have the results of the trials of three small vessels fitted with geared turbines, but, at that time, these trials were incomplete.

The vessels are named *Curzon*, *Elgin*, and *Hardinge*, after three Viceroys of India, and are intended to connect Ceylon with the Indian mainland by a short sea passage in smooth water instead of the long and occasionally disagreeable voyage which hitherto has been the only available means of transit.

The South Indian Railway Company, owners of the vessels, applied for guidance to the late Sir William H. White, who furnished them with an outline design of the internal arrangements and a specification covering all the requirements of the service, at the same time leaving the builder a very free hand as to the form of the vessel, scantlings of material, power of propelling machinery, and other details.

Sir William White's foresight was evinced by the fact that the whole contract has been completed without any alterations on the working plans or any extra charges of importance.

The vessels are 250 ft. long by 38 ft. broad, and displace about 865 tons on 6 ft. draught of water.

The principal conditions of the contract were that the vessels should carry 160 tons of dead weight on a draught of water not exceeding 6 ft., and, while so loaded, be capable of steaming 20 sea miles at the rate of 16½ knots, starting from rest, no allowance being made for the time occupied in getting under way. The fuel used on trials was to be as nearly as possible equivalent in heating power to the Indian coal available on service, a sample analysis being furnished which showed 63·26 per cent. of fixed carbon. The coal actually used contained 63·86 per cent. of fixed carbon, being a rather poor

sample of Scotch coal, but almost exactly what was wanted. Yarrow boilers were fitted, the heating surface being 7,000 sq. ft., and the grate 168 sq. ft. The boilers were arranged to be fired with oil if that should be found advantageous.

It will be observed that the actual full speed to be attained was not stated in the conditions, nor were any data available whereby it might be computed. It was considered safe to assume 17 knots, and estimates were made on that basis, as, if 17 knots could be maintained for 19 miles, there would be left one mile of distance and over five minutes of time to attain full speed. This was deemed to be sufficient.

A preliminary trial of the *Curzon* was made, on December 10th last, to test the working of the turbines, and, the weather being very calm, advantage was taken of the divisions on the measured mile in the Gareloch to learn something about the acceleration of a vessel starting from rest. The results are shown on the diagrams.

Diagram No. 1 (Plate XV.) shows, on a base of one nautical mile, the time for each fraction of a mile and the increasing speed during acceleration.

Diagram No. 2 shows, on a base of speed in nautical miles per hour, the revolutions of propellers, the slip in knots and the slip per cent. during acceleration.

Diagram No. 3 shows the revolutions, slip and shaft horse-power during a progressive trial of the *Elgin*, also the effective horse-power (which Messrs. Denny were so kind as to ascertain for us in their experimental tank), and the ratio of effective to shaft horse-power.

The dotted curves on No. 1 show the computed speed during acceleration, also the time required to run one mile starting from rest. These were constructed by Mr. Mumford, superintendent at Messrs. Denny's experimental tank, from calculations based on the resistance of the model. The agreement between the curves constructed from calculations made on a model and those from experiment on a full-sized ship is extremely close. If Mr. Mumford is present I hope he will tell us how this clever piece of work was done.

The 20 miles distance, starting from rest, was steamed by the *Curzon* in 65 minutes, equal to a mean speed of 18·46 knots, the tide being with her for the first half and against her for the second half, the wind fresh, about three points abaft the beam.

The time occupied by the *Hardinge* was 66½ minutes, equal to 18·045 knots, she having a stiff breeze against her and a weak tide in her favour, high water at Greenock being about 8 o'clock on that day.

It was not considered necessary to repeat this trial with the third vessel, and instead of this a progressive trial was made on the measured mile, and a run of two hours at high speed was utilised in measuring the water consumption.

During the latter the revolutions were somewhat reduced by a strong head wind, the average being 482·8 per minute, corresponding to a speed of 17·8 knots.

The mean shaft horse-power was 2,390 and the consumption of steam, for turbines only, 12·55 lbs. per hour per shaft horse-power.

The consumption of steam by the auxiliaries amounted to nearly 3 lbs. per hour per shaft horse-power of main engines. Among the auxiliaries are reckoned the engine driving a large dynamo, and the steam steering gear. The temperature of the feed was 205° Fahrenheit. The air pressure in the stokehold did not exceed a quarter of an inch of water column.

An attempt was made to record the noise of the gearing on a new phonograph obtained for the purpose, but, although the sound of the engine-room telegraph gong and some vocal efforts were clearly rendered by the instrument, the sound of the gearing could not be recognised amidst the scraping of the reproducer on the wax cylinder.

Professor Biles (now Sir John Biles), in his interesting remarks on Sir Charles Parsons' paper, quotes a shipowning friend repelling the suggestion to permit the innovation of "cog-wheels" in a passenger steamer's propelling machinery.

The use of cog-wheels cannot be regarded as a novelty even in high-class ocean-going passenger steamers. I can remember the building, in 1855, of the steamer *Oneida*, which was then deemed large, seeing that only 25 steamers of the same or greater register tonnage were on the Mercantile Navy List at that date. This vessel was 307 ft. long by 39 ft. beam. Her gross tonnage was 2,284, and her net register 1,372 tons. Her engines were direct-acting, but the results were so unsatisfactory that, before she was two years old, that is in 1857, she was put into our hands to undergo certain alterations and receive new geared engines of about 3,000 horse-power. These engines, cog-wheels and all, appeared to work satisfactorily for 17 years, though I fear they could not have been remarkable for economy of steam, and in 1875 she was converted into a sailing ship. Thirty years later, being then 50 years old, she disappeared from the British register, having found a new owner in Chili.

High speed of piston being somewhat of a bugbear to engineers in the fifties, geared engines were common enough in channel and coasting steamers, and even Robertson's frictional gearing, where the wheels and pinions had grooves in place of teeth, was not unknown in marine work.

What was tolerated 60 years ago, with cast-iron pinions and mortise wheels, might surely be adopted now with the refined methods of gear-cutting which Sir Charles Parsons and others have made available.

DISCUSSION.

Professor Sir JOHN BILES, LL.D., D.Sc. (Vice-President): Mr. Chairman, Ladies and Gentlemen, I have heard only part of this paper read, but I think you will agree that it is a paper which does not require a large amount of intelligence to appreciate what there is in it. It is a record of a very satisfactory performance in the latest development of marine propulsion. I am sure you will all be willing to congratulate Dr. Inglis on the success of the work done by his firm. If he could convey to us a full impression of the amount of noise that the gear makes in the ship he would convey some additional valuable information. I take it that at the present moment nobody disputes the mechanical advantage of geared turbines. There are still some people who have their doubts on the question of the amount of noise that the gear makes, and whether it is a permissible amount. It is exceedingly difficult, of course, to convey to anyone what a noise really means, and only individuals themselves can judge by passing from the case of one steamer to that of the next. We had a most pleasant and delightful trip yesterday, and in that trip we appreciated some of the small noises that took place in that particular ship, and those who notice these things were able to compare to their own satisfaction the relative amount of noise in that ship with another one, but it is practically impossible to convey that impression to other people. Until we get, as Sir Archibald Denny suggested in our last discussion on this subject, some measure of noise, it is very difficult to arrive at definite conclusions, though I think it is agreed that the important question in connection with geared turbines at the present moment is whether or not the noise is one which can be tolerated. Now there are a great many noises in ships without geared turbines, and, in my judgment, there are noises that are tolerated and almost pass unnoticed, which are much more objectionable than the new noise of the gears. Perhaps Dr. Inglis, who has had full experience of these ships, will be able to convey to us his impression of the relative noise in them compared with that of an ordinary steamer.

The problem that is laid down here is apparently a very unusual one, that of obtaining a speed which was not stated, but which was necessary to cover a certain distance in a certain time starting from rest, and, as Dr. Inglis says, there were not any data available whereby the actual full speed could be computed. I think that is true, as far as I know, if one bases it upon actual ship results, but I think it was possible to estimate it, at the time Dr. Inglis considered this inquiry. If he had happened to be a student of mine in the University class, he would have had to go through an example there, whereby, given a curve of resistance and a known amount of thrust, or a calculable amount of thrust, it would be possible to very closely approximate to the time required to arrive at a definite speed in a ship form, and also to the mean speed over the time of acceleration. That method was made use of in the case of the *Olympic* and *Hawke* collision, where the *Olympic* was accelerated and retarded during the critical part of the period during which she was likely to come into collision, and a very complete calculation was made showing the variation of speed in terms of thrust and time, based upon the curve of resistance of the *Olympic's* form. In the book which I published some time ago, you will see a method laid down in the chapter on "Resistance," and I should be glad to know how the results compare with those given in the paper. Perhaps Mr. Mumford or Sir Archibald Denny will tell us the method by which they were arrived at. It must obviously be a basis of curve of resistance, and here we have, I think, for the first time, a relation between the calculation based on the curve of resistance and thrust, and the actual results in the ship, so that you get what is of infinitely more value than a class-room exercise, viz., the actual results of a full-sized ship. I think we are very much indebted to Dr. Inglis for having carried out these experiments and for having given us the benefit of them.

Sir Archibald Denny, Bart., LL.D. (Vice-President): I think, Sir, that I will not discuss this paper, but I will ask Mr. Mumford to do so, a he will explain his work far better than I can. I should like, however, to congratulate Dr. Inglis on the success of his work, and also upon the full account he has given us, because the more data we have of speed, power and revolutions, and so on, the better for our work. He has, however, missed out an important thing, and, like Oliver Twist, who always asked for more, I will ask him why should he not give us the dimensions and the particulars of the propellers ?

Mr. E. R. Mumford (Member): Mr. Chairman, Ladies and Gentlemen, before giving the explanation which Dr. Inglis has invited me to make, I desire to take this opportunity of congratulating him and his firm on the successful fulfilment, by the three vessels, of the rather unique speed conditions that were specified and guaranteed. In the racing world a standing start is quite usual, but the speed of power-driven ships is generally specified for a "flying start" over a measured distance. There is frequently inserted in the specification, however, more particularly for turbine-driven vessels, a clause defining the distance in which a vessel must be brought to rest from a certain speed. A proper understanding and solution of the problem of bringing a vessel to rest involves a consideration of the same factors as the problem which was set to Dr. Inglis of accelerating from rest. These factors are the total mass dealt with, the resistance at all speeds, and the thrust of the propellers at all speeds. The mass affected is that of the vessel plus some of the surrounding water. Froude's finding, from the *Greyhound* experiments, was that the combined mass of vessel and water is 20 per cent. greater than that of the vessel, and this is usually accepted and found to give accurate results. The resistance of the vessel at all speeds up to maximum speed was in the present case deduced from experiments with the model in Messrs. Denny's experimental tank. The model was tried with and without shaft brackets, and the resistance which has to be overcome by the thrust of the propellers is the resistance of the ship with shaft brackets, augmented about 10 per cent. by the action of the propellers.

The thrust of the propellers during acceleration from rest is not amenable to such accurate determination. Messrs. Denny have made, however, numerous retardation tests with full-sized vessels, by an accurate method (which I need not now describe) which gives a continuous record of the rate of retardation, and the revolutions per minute of the propellers, during retardation from full speed to rest. These records permitted of an estimate of the revolutions per minute during acceleration of the three vessels built by Messrs. Inglis, and from the revolution curve thus deduced (see next page) it was possible to estimate the increasing thrust of the propellers during acceleration with sufficient accuracy, making a slight allowance for the time taken to turn on full steam. The thrust at full speed must, of course, balance the resistance at full speed, while the thrust at zero speed was determined from the results of experiments with model screw propellers, assuming a diameter and pitch of propellers of 4 ft. 8 in. The thrust is a function of $D^4 \times (\text{Revs. per min.})^2$ for no speed of advance of ship. Thus, the thrusts for zero speed and full speed being determined, it only remained to connect the two points by a curve of parabolic character, slightly altered in curvature at the low-speed end, to allow for the time taken to turn on full steam.

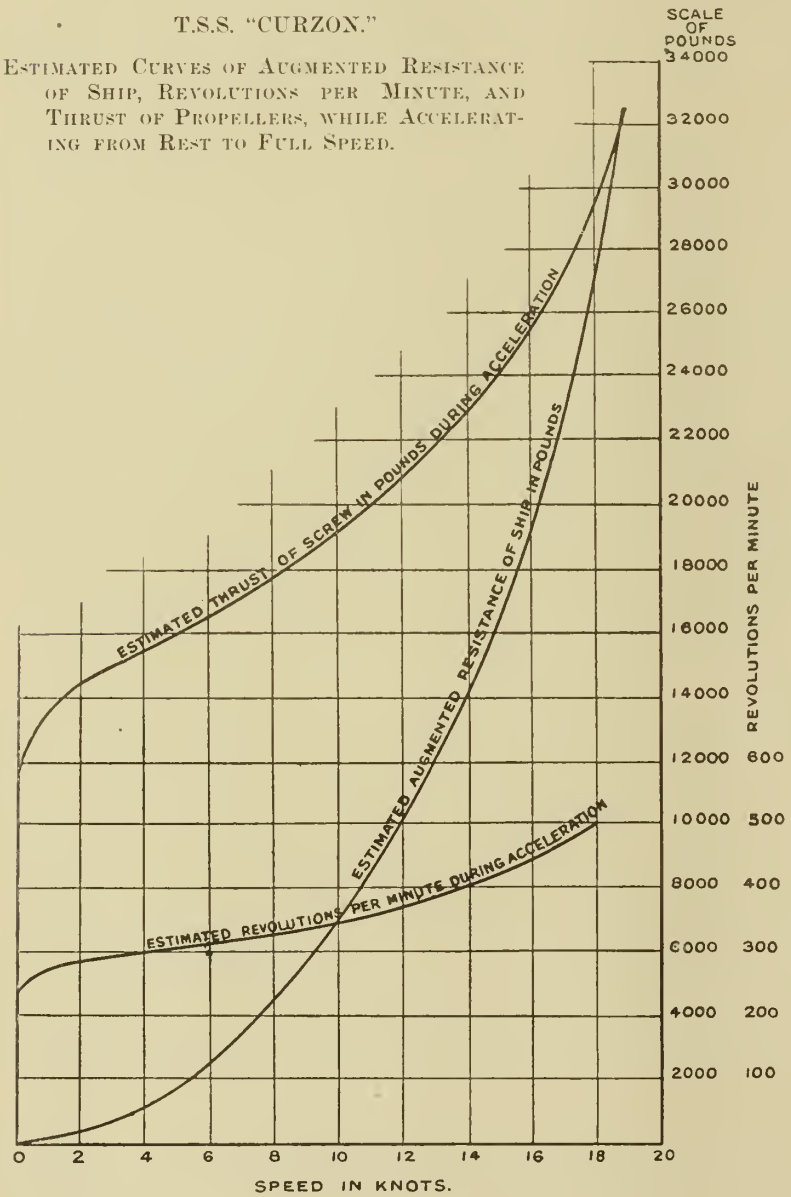
Having thus obtained all the factors required for determining the rate of acceleration, it only remained to apply, by the method of graphical differentiation, the average accelerating force—*i.e.*, the average thrust minus the average resistance—during each small increment of speed, to a mass 20 per cent. greater than the mass of the vessel. In this way the time, distance, and average speed during each increment was calculated, and the integration of the whole gave the dotted curves on Fig. 1 (Plate XV.). I should add that when the calculation was made the weight of the ship was taken to be 800 tons, and

that the actual displacement of the ship on trial was 847 tons. The effective horse-power given by Dr. Inglis in Fig. 3 (Plate XV.) is for the ship fitted with shaft brackets.

Before I sit down I should like to take this opportunity of saying that many of the published results of stopping trials of vessels—*i.e.*, the distances in which vessels have been brought to rest—are demonstrably impossible. Such results are frequently quoted as authoritative, and do harm in many ways, but the method I have described, of estimating the rate of acceleration, can be applied to estimate the rate of retardation with sufficient accuracy to provide a healthy check upon all such published figures, some of which are very misleading and harmful to young students and others, and cause shipowners to specify impossible conditions, which the shipbuilder must either evade, or pay the penalty for non-fulfilment.

T.S.S. "CURZON."

Estimated Curves of Augmented Resistance of Ship, Revolutions per Minute, and Thrust of Propellers, while Accelerating from Rest to Full Speed.



Mr. W. Brooks Sayers: Mr. Chairman and Gentlemen, I am greatly interested in the subject of this paper, inasmuch as some years ago, when Mr. Mavor first brought his paper upon electric transmission before the Institution of Engineers and Shipbuilders, I exhibited a model gear which I thought had great advantages over the direct form of gear. The question of noise is put as one of chief importance, and I should like to say a word about it. It is quite obvious, I think, that the question of noise depends upon two factors: one, the accuracy with which the gear is cut, and the other the resilience or the elasticity of the material, because these two factors determine the violence of the collision. Supposing a gear is perfectly cut, even then the steel is not absolutely rigid, and the pair of teeth that are in heavy compression yield to a certain extent, with the result that the next pair come more or less into collision. Now

that is greatly reduced, of course, by the device of making the teeth helical. They roll into contact more or less. But there is another method by which collision is very largely influenced, and that is by adopting, instead of ordinary external wheels, an epicyclic form, and I exhibited a model of such a form which I named "circle-epicyclic."* I think it is of the utmost importance at the present day that the question of the form of gear should be looked into more closely. An epicyclic gear can be made use of to change speed in large ratios without occupying immense space as regards diameters, and the magnitude of the collision, and consequent shock and noise, can be reduced to practically nothing, because the wheels in mesh roll together.

Mr. Max Wurl (Member): Mr. Chairman and Gentlemen, as the question of steam consumption is of special interest in connection with geared turbines, I would like to ask Dr. Inglis to give us the method by which the figures on page 139 have been obtained, and to state the vacuum at which the turbines were working at the time, in order to enable us to compare this with other systems of propulsion. The consumption of 12½ lbs. seems a very good result for the size of turbines, but the 3 lbs. for auxiliaries is rather higher than what one would have expected.

Mr. S. J. Reed:† Mr. Chairman and Gentlemen, the results obtained with the three ships built by Dr. Inglis' firm are very gratifying, and the addition of these vessels to the fleet of steamers in which transmission gear is employed between the turbines and propellers is very welcome. I have been very closely associated with Sir John Biles in connection with the fitting of this form of machinery to the London and South Western Railway's steamers, the *Normannia* and *Hantonia*, and Messrs. Inglis' ships, although of smaller power, had the same general arrangement of machinery. The good results obtained with the two South Western Railway ships on trials have been fully maintained since they have been on regular service, which now amounts to about 16 months. I am speaking from experience, when I assure you that the Southampton-Havre service is no light one, yet there is no sign of any pronounced wear on the gears, and there have been no troubles at all with the machinery. The coal consumption has been carefully kept by Mr. Dashper, the Marine Superintendent Engineer to the Company. During the first six months these ships consumed about 43 tons of coal per double trip, with a service speed of approximately 18 knots; this speed on service is equivalent to very nearly 19 knots under trial trip conditions. The reciprocating engined ships on the same service, when running at a similar speed, consumed about 44 tons of coal. The approximate displacement of the geared turbine ships is, however, 1,900 tons, while that of the reciprocating ships is only about 1,600 tons. The displacements given are only approximate, as they vary a good deal, depending on the amount of cargo that the ships carry on their trip. Assuming that power and displacement vary directly, which, with these particular ships, may be taken as somewhere nearly approaching the truth, the advantage in favour of the geared turbine ships was about 18 per cent. The reciprocating ships against which the comparisons had been made were fitted with two sets of 4-cylinder triple-expansion engines, and may be taken as having a fair average consumption. The coal used includes the amount consumed in port, which is a considerable proportion of the total. In the case of the geared ships this port coal has been kept separate; however, the detailed figures are not of much interest at the moment, as it is the total coal that the shipowner has to pay for. Further, in the case of the reciprocating ships the port coal was not separated, and the figures are, therefore, not available for comparative purposes.

* A description and illustrations of this form of gear will be found in the records of the discussion of Mr. Mavor's paper in the Proceedings of the Institution of Engineers and Shipbuilders in Scotland. (Vol. LI. 1907-8.)

† Written contribution to the discussion.

The manner in which the coal consumption has been obtained with these ships is to weigh into the bunkers each time the ship is coaled, the bunkers being always filled. With this service the ships are coaled each time they arrive in Southampton, which, excluding Monday morning, is every second day, and an average is made of the weights for the month. When these geared turbine ships are on a 19-knot service speed the average coal per double trip is approximately 47 tons. The coal consumption of these ships up to the present time is for all practical purposes the same as I have already given, although during some of the months of this year it has been rather less. With these two ships it is remarkable how close the coal consumptions are, the maximum variation between them, now that they have settled down to their service, being not more than 3 or 4 cwt. for the month.

I should like to ask Dr. Inglis how the water consumption of the *Elgin* was obtained. In the case of the South Western Railway boats, it was measured by means of the feed pumps, and while the measure was being taken no supplementary or auxiliary feeds were in use, so that the feed pump was simply returning to the boilers the amount of water discharged by the air pumps. The auxiliary consumption was obtained in the same manner after the main turbines were stopped, and this, in the case of the *Normannia*, deducted from the over-all consumption, gave for the turbines alone a consumption of 12·2 lbs. I am very glad that Dr. Inglis has attempted to measure the noise of the gear by means of a mechanical instrument, and am only sorry it did not prove satisfactory. I think that the judging of the noise of the gears is one of the most deceptive things imaginable ; some days they appear to be much quieter than others. However, I am quite satisfied that with the Southampton boats the noise has lessened since they have been on service, especially when going astern.

In the case of Dr. Inglis' ships it would have been a very difficult matter to obtain a reasonable consumption with direct-coupled turbines, while in the case of the Southampton boats it would have been a difficult matter to actually beat the reciprocating engine. When it came to the higher speed and powered ships, however, a far better economy could be obtained with turbines than with reciprocating engines. When considering transmission with these latter ships the question of economy has only to be considered in conjunction with direct-coupled turbines. Judging from the economy obtained with the high-powered land turbines, there would be very nearly as much gained by the adoption of transmission with this latter class of ship as with the slower type. Another advantage of transmission was that superheat could be more safely applied to the comparatively small high speed turbines. In fact, they were in exactly the same position as land turbines, and there seemed no reason why superheating should not be introduced for marine work. With the smaller powered ships mechanical transmission gear appears to have it all its own way, but it is a little doubtful at present what is the upper limit to the power to which it can be applied in practice. Unless the tooth speed and bearing pressure between them are very much increased above that adopted at present, the gears will have to be of very large size when it comes to big powers. This could be got over by dividing the power between three or four shafts, but I think most engineers are inclined to return to the simplicity of the two-shaft arrangement if possible.

Mr. John Inglis, LL.D. (Vice-President) : Mr. Chairman and Gentlemen, with regard to Sir John Biles' remarks on noise, we made several attempts, as I have said in the paper, to record this on a phonograph, but although we could easily record the noise of a boiler-shop, or a loud-speaking voice, it was impossible to get any record which we could bring into this room and reproduce for the benefit of the audience. You may judge from this that the noise was less objectionable than one might expect. These vessels had above the main deck a flying deck on which there were a chart-house and a companion-house, and in those houses on the upper deck the noise was almost indistinguishable. It was loud enough

in the engine-room, certainly, but so are the noises in most engine-rooms, and I should say there was nothing objectionable in the noise we experienced, even for a passenger steamer.

I am obliged to Mr. Mumford for his extremely interesting remarks; in fact, when I proposed to read this paper, I had a notion that, like Falstaff, I might be the cause of wit in other men if I had none of my own, and I think I am entitled to the gratitude of the meeting for having called forth Mr. Mumford's remarks. The consumption of water, I may say, was taken by actual measurement while the vessel was steaming on the Firth of Clyde. The consumption of the auxiliaries alone was measured while the vessel was at our wharf. The electrical installation was a very large one, the steam winches and capstan being driven electrically. The same dynamo also drives the electric light, and there was in addition a subsidiary dynamo for that purpose. The feed make-up was included in the consumption of water by the auxiliaries.

The Chairman (Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland): Gentlemen, I am sure I express your wish when I ask you to accord a hearty vote of thanks to Dr. Inglis, not only for his exceedingly interesting and valuable paper, but also for the additional information which he has so quietly and unobtrusively embodied in his reply to the discussion.

A DEVICE TO FACILITATE THE COUPLING OF CRUISING TURBINES.

By Harold E. Yarrow, Esq., Associate-Member.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913; Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland, in the Chair.]

Owing to the uneconomical performance of turbines when developing powers considerably below full power, attention has recently been directed to improving the economy of vessels when cruising at low speeds by the introduction of special motors such as turbines, internal combustion engines, or reciprocating steam engines.

Sir Charles Parsons has already constructed successfully turbine installations having special cruising turbines. Certain difficulties that arise from the use of an additional motor, transmitting its power simultaneously with the main turbine, may render it necessary to use the main turbine and the cruising turbine independently of one another, the vessel being driven by the one or the other, and not by the two in combination. This involves connecting and disconnecting the low-power motor from the main shaft.

The cruising turbine or engine may drive the main shaft through the medium of electrical or hydraulic transmission, or by simply throwing a clutch in and out of gear; the latter arrangement undoubtedly offers advantages, there being no appreciable loss of power, and the increased complication and cost of an electrical or hydraulic transmission are avoided.

The use of a friction clutch on board ship may lead to difficulties, especially in view of the comparatively large powers that have to be transmitted. The simpler and more direct method is by throwing a dog clutch in and out of gear, so that the cruising motor can be coupled up or not as desired.

This paper is not intended to deal with the design of a suitable dog clutch, but it has been prepared with a view to submitting to the Meeting devices which will facilitate the coupling of the clutches.

When a high speed is demanded it is contemplated that the main turbine only will be working, and when a cruising speed is required the cruising motor only will be working, the main turbine in this case being simply driven round by the cruising motor. If the cruising motor consists of a steam turbine, the exhaust from it could of course be utilised in the main turbines to assist in the propulsion of the vessel.

Fig. 1 (Plate XVI.) represents one arrangement of turbine machinery.

Assuming that a vessel is being propelled at low speed by means of the cruising motor, which would probably convey its power to the main shaft through gearing, and it is desired to increase the speed and propel the vessel by the main turbine, no serious difficulty would be found in throwing the cruising motor out of gear by means of an ordinary well-designed clutch. It is, however, not such an easy matter to make the change from the full-power turbine to the low-power motor, because in this case the clutch has to be put into gear. This can be done by stopping the vessel, but that may not always be permissible, and if it can be avoided it is certainly preferable.

On account of the large powers now in use even when cruising, if the two shafts that it is desired to couple together are not rotating at nearly the same speed, it would not be possible to engage the clutch without involving an objectionable shock. On the other hand, if the clutch is put into gear when the shafts are revolving at approximately the same number of revolutions, it is evident no appreciable shock would be caused. It therefore becomes necessary to adopt some device by which the relative speeds of the two shafts can be ascertained; and when they are revolving at practically similar speeds the clutch could be put into gear, and the shafts then connected together.

Fig. 2 shows one arrangement proposed, from which it will be seen that on the outside of each coupling a screw thread is cut, the threads on both couplings being similar. By watching the thread when the couplings revolve it makes their relative speeds at once apparent, and when the screw threads appear continuous the two couplings will be revolving at equal speeds. If they are rotating at different speeds the one which is revolving the more rapidly will make the screw thread appear to to be travelling faster. The optical effect produced by the rotation of the clutches upon which the screw thread is cut makes the progression of the thread or spiral line in the direction of the axis of rotation readily apparent, the eye at once recognising any difference in speed between the two shafts.

Fig. 3 shows another method of carrying out the same idea. It will be seen that two pulleys are driven from each of the two shafts that it is required to couple together, the pulleys revolving in opposite directions. Each pulley would be attached to a short spindle having at its extremity a bevel wheel gearing in with another bevel wheel, which latter would be fixed in a revolving frame, and therefore forming a differential gear similar to that used in a motor car. If the two shafts are revolving at the same speed it is evident that the revolving frame will remain stationary, but if there is a difference in the revolutions of the two shafts the difference will make itself apparent by causing the revolving frame to rotate either in the one direction or the other, depending upon which shaft is travelling faster. On the outside of the

revolving frame it will be seen a screw thread is cut so as to make apparent the speed and direction of rotation of the revolving frame. When the speeds of the two shafts approximate very closely, the revolving frame will almost stop, and the coupling can be connected with the minimum shock.

Fig. 4 (Plate XVI.) represents another device by which a pointer indicates the relative speeds of the two shafts.

Many other devices could be designed for effecting the same object, but it will be admitted that without some means of this kind it would be very difficult to form a correct estimate of the speed of rotation of the two shafts in line with one another.

Although such a device has not been fitted by us in any vessels, from trials made in our shops it has been found very easy to determine the proper moment to engage the clutch.

With reference to the coupling itself, in which an ordinary dog clutch is used, it is desirable that the two clutches should only be capable of being put into gear, so that the same faces are always in contact with one another, because with the greatest possible accuracy in workmanship it may be found that the work will not be taken up equally by the different surfaces in contact, and the faces of the clutch will readily wear. Therefore, it is submitted that whatever clutch is adopted the same faces should always be in contact, and this can be effected by the projections of the dog clutch being of unequal size.

Another point which should be borne in mind in connection with such clutches is to design the bearings which support the shafts close up to the clutches, so that they should not be subjected to unequal wear. This is necessary, because if there is a difference in the amount of wear to which the two bearings are subjected, causing the shafts to get out of line, the want of alignment thus arising between the two clutches will cause movement between them and lead to trouble. For this reason it may be desirable that a certain length of shaft should be interposed between where the power is transmitted and the clutch, which would allow for the unequal wear of the main engine or turbine bearings; while the special bearings which have only to support the weight of the shaft near the clutches may be considered as likely to wear evenly.

The devices which I have had the privilege of describing are, in a great measure, due to my Father, to Mr. Marriner, and Mr. Cotton, to whom I am further indebted for their help in the preparation of this paper.

A demonstration on a large scale model of the coupling gear was given at the conclusion of the reading of the paper.

DISCUSSION.

Mr. A. F. Yarrow (Vice-President): Mr. Chairman and Gentlemen, I should like to point out that in the model there is a break in the spiral line owing to its being continued on the projections of the clutch, and the effect is not quite so pronounced as in practice, when there is no break in the line. In that case the spiral thread looks quite continuous, while in the model, owing to the line being confused by the projections, the apparent movement of the line parallel to the axis is not so pronounced.

Mr. J. S. Nicholson, B.Sc.: Mr. Chairman and Gentlemen, it seems to me that a very simple method of determining the relative speeds of the two shafts would be by some stroboscopic means. Last month I had the pleasure of reading before the Institution of Engineers and Shipbuilders a paper on "The Stroboscope in Speed Measurements" by Professor Robertson, of Bristol, and I showed a number of experiments illustrating the manner of applying this stroboscopic method. In this case all that would be necessary would be to have a couple of discs, one on each half of the coupling, with a certain number of slots in the one and the same number of spots in the other. On looking through the slots of the one disc at the spots on the other we can tell at once when the two halves of the coupling are running at the same speed, as the spots then appear to be stationary. If the difference in speed is small the spots will appear to rotate in one direction or the other, the direction depending on which half of the coupling is the faster. The two halves, however, might be running at just the same speed but not quite in phase with each other ready for coupling. In that case, in order to determine when the projections on one half of the coupling are opposite the recesses on the other half it is only necessary to fix the line of observation. Any difficulty in observing the discs on the coupling may be obviated by some optical method, or the discs may be mounted on side shafts driven by means of chains from the two halves of the coupling.

Engineer-Commander Percy Stocker, R.N.: Mr. Chairman and Gentlemen, in the case of internal combustion engines there is one point which, if the subject of clutching up is going to be seriously considered, must be carried out, and that is that the two shafts must form the equivalent of a solid shaft. The noises which we have been hearing about in the last lecture are really deafening, and the design of clutch must be made so that the clutch is locked up and runs absolutely without a sound. Being in the Submarine Service, my experiences are confidential, but I think I can make that point without disclosing any secrets. I have had about three years' experience of these clutches, and the conclusion we have come to is that unless you have the equivalent of a solid shaft no clutch is very satisfactory.

Sir Archibald Denny: Can you do it?

Engineer-Commander Stocker: Yes, it is done.

Mr. W. H. Whiting (Member of Council): Mr. Chairman and Gentlemen, I do not intend to take part in the discussion as regards the details of these devices, but as it is particularly for the Admiralty that Messrs. Yarrow are working in this matter, and as it is we who really make the demands on them in regard to these great variations of speed, I think we are extremely indebted to Messrs. Yarrow for the way in which they are setting themselves to get over the practical difficulties involved, and I hope that we shall before long have an opportunity of seeing how these proposals are applied in practice, and with what success.

Mr. Harold E. Yarrow (Associate-Member): Mr. Chairman and Gentlemen, I do not think that any questions arise out of the discussion. Mr. Nicholson's ingenious device with slots in the couplings to determine when they are revolving at similar speeds is most interesting, but it must be remembered that not only is it necessary to determine when the shafts are revolving at the same speed, but it is also required to determine their relative speeds, so that they may be easily regulated to revolve at the same number of revolutions when it is required to couple them together. I would like to thank Mr. Whiting for his kind remarks, and also the meeting for the way in which they have received my paper.

The Chairman: Gentlemen, it is not a mere formal vote of thanks that I ask you to render to Mr. Yarrow. We are indebted to Messrs. Yarrow for many things, and not the least of these is the free use they give us of their information.

PERFORMANCE ON SERVICE OF THE MOTOR-SHIP "SUECIA."

By I. Knudsen, Esq.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913; Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland, in the Chair.]

In the autumn of 1911 the Rederiaktiebolaget Nordstjernan Company, of Stockholm, ordered six motor liners from Messrs. Burmeister & Wain, the firm with which I am connected, the dimensions of the vessels being as follows:—Length, 362 ft.; breadth, 51 ft. 3 in.; and depth, 34 ft.; carrying capacity 6,500 tons deadweight.

The machinery consists of two main engines, each of 1,000 I.H.P. (4-cycle Diesel engines with 8 cylinders), besides 2 auxiliary Diesel engines each of 200 effective horse-power for working the compressors, auxiliary machinery (such as winches and steering gear), and for the production of the electric light; the machinery is in other respects similar to that of the *Selandia*, which was described by the Author in a paper read before this Institution last year.*

As to passenger accommodation, there are only eight cabins for first-class passengers, which is much less than is provided for in the *Selandia;* the cabins, however, are roomy and modern, and provided with bath and toilet rooms. Further, there is ample saloon accommodation, and a hospital. The *Suecia* is intended for the Sweden to La Plata service, and she will be chiefly employed for cargo purposes, being fitted with the most modern loading and discharging gear, such as double derricks and double winches.

The vessel was launched on November 2, 1912, and the trial trip took place on December 17, 1912. The next day she went to Limhamn, where she took in a cargo of 15,000 barrels of cement, then she sailed to Stockholm, where—as far as I know—a cargo of 4,000 tons was taken in. On December 23 she left for Gothenburg, and here she took in still more cargo, and left on December 31, arriving at Christiania on January 1, where her cargo was completed, and on January 4 a trial trip took place in the Christiania Fjord with the vessel fully laden. When the trial trip had been

* Trans. I.N.A., Vol. LIV. p. 68.

completed, she went to London, where she arrived on January 9, and went from London to Rio Janeiro, arriving at that port on February 1.

I will refer to certain matters concerning the *Suecia* owing to some remarks made in the discussion of the paper which was read before the Institution of Naval Architects last year, to the effect that no information was given with regard to the efficiency of the propellers, &c., compared with that of ordinary steamers. The reason why no such information was given in connection with the *Selandia* was, that with this vessel, no trial trip took place with the ship fully laden, and consequently we possessed no comparative results. With the *Suecia*, however, we succeeded in carrying out this trial trip in Christiania Fjord, and I am glad to have this opportunity of publishing the results.

The full lines on Fig. 1 (Plate XVII.) give the curves showing the indicated horse-power, the number of revolutions, and the speed obtained with the *Suecia*.

The dotted lines on this figure give similar particulars obtained with the *Princessan Ingeborg*, a vessel built a few years ago for the same owners, but fitted with steam engines.

As appears from the dimensions given on these diagrams, the vessels are almost alike, both in dimensions and also with regard to the fineness of lines. If the two vessels are compared, it will be seen that to obtain the same speed for the same displacement the same indicated horse-power is required, whether it is to be used in the motor liner or in the steamer. According to this result, there seems to be no difference in the power required to propel a vessel, whether this is effected by means of a single, large, slow-running propeller, as in the steamer, or by two relatively fast-running small propellers, as in the motor liner. The propeller in the steamer has a diameter of 17 ft. 6 in., and in the motor liner of 10 ft.

In this respect it must be remembered that in the *Suecia* the main Diesel engines do not work the pumps, nor the two first stages of the air compressors. The relation of the indicated horse-power to the brake horse-power of the engines will, therefore, be about the same as in a steam engine, and the combined efficiency of the mechanism of the engines and of the propellers is about the same as obtains with the steam engine.

The weight of steam installation in the *Princessan Ingeborg* brought up to correspond with the H.P. of the motor installation amounts to 570 tons, as an extra main boiler was fitted in view of the cleaning of the boilers. The total machinery in the motor ship *Suecia* weighs 470 tons.

With regard to the space occupied by the engines in the *Suecia*, this is 41 ft. of the length of the vessel, whereas it is 66 ft. in the *Princessan Ingeborg*, although the H.P. as stated above is less in this vessel.

It should be stated that the trial trip took place under good conditions, such as fine weather and deep water. The vessel, however, has now returned from her maiden voyage, and from the speed attained on the Atlantic, the comparative trial trip results seem to hold true in the practical performance of both vessels.

In the trial trip in Christiania Fjord the oil consumption per indicated horse-power hour was measured, and a result was obtained which no doubt is the best hitherto obtained, viz., 134 grams of oil per indicated horse-power hour, measured on the main Diesel engines, and when everything is allowed for, such as the consumption of the auxiliary engines, &c., the oil consumption per indicated horse-power of the main Diesel engines amounts to 154 grams, including the fuel oil necessary for the working of the auxiliary engines, steering engines, pumps, &c., but exclusive of the oil used for heating the vessel.

The mechanical efficiency of Diesel engines of the type used, but with the whole of the compression of the air necessary for fuel injection performed by the engines themselves instead of by auxiliaries, has been shown by bench trials to be 80 per cent., so that the oil consumption per brake horse-power of such a marine Diesel engine would amount to 167½ grams per brake horse-power.

The diagram (Fig. A), which was taken during the trial trip, shows by its shape and appearance that the engine is extremely economical.

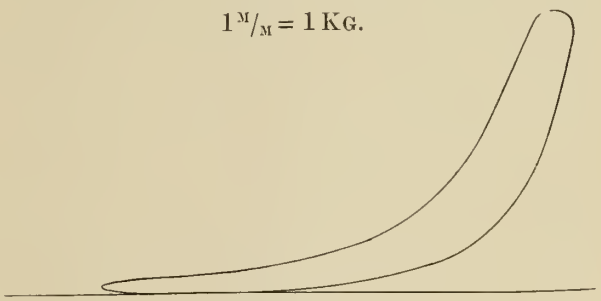


FIG. A.

The *Siam*, which we recently built for the East Asiatic Company, was, until now, the biggest sea-going motor liner, her length being 410 ft., her carrying capacity 9,200 tons, and the horse-power 3,000.

It may, perhaps, be of interest to follow the development of motor liners, and I may add, therefore, that at present the following vessels have been ordered from my firm besides those already delivered :—

One motor liner similar to the *Siam*, 410 feet long with engines of 3,000 H.P., for the East Asiatic Company.

Five motor liners referred to above, similar to the *Suecia*, for the Rederiaktiebolaget Nordstjernan.

One vessel, 405 ft. long, carrying capacity 7,500 tons, with engines of 2,500 H.P., for the United Steamship Company of Copenhagen.

To replace the *Fionia*, which was sold to the Hamburg-America Line, we are constructing for the East Asiatic Company a motor liner, 390 feet long, with engines of 4,000 H.P. This company has also ordered from us four other vessels, each with a length of 410 feet with engines of 4,000 H.P.

All the figures given of the horse-power are exclusive of the auxiliary Diesel engines.

DISCUSSION.

The CHAIRMAN (Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland): Gentlemen, as I think I may have been one of the guilty ones who asked for more information regarding propeller efficiency, or rather propulsive efficiency, in the case of the Diesel-engined ships, I now desire to thank Mr. Knudsen very heartily for the information which he has given us regarding the results of those two similar vessels. I do not know that this quite clears up the whole matter. I am still of the opinion that it is difficult to obtain a high propulsive efficiency in an ordinary cargo vessel with a propeller revolving at the speeds which we know prevail in the *Selandia* and similar ships.

I have been associated with Diesel marine engine work recently in the case of the M.S. *Eavestone*. In this vessel it was at first intended that the speed of revolution of the engine should be 115, and a propeller was designed for that speed, but it was never fitted. The propeller which was fitted in the first instance was designed for 105 to 110 revolutions, and the speed of the vessel obtained with that propeller was very unsatisfactory indeed in relation to the power the engine was developing. After considering the matter we decided to fit a propeller of the same diameter as the first, but of increased pitch and surface, with the result that the speed of the ship was increased by quite 12½ per cent., while the revolutions fell to 93 or 94 on trial and about 90 at sea. I think that with the latest propeller we are getting as much effective work per brake horse-power as we would have done with a steam engine. In comparing Diesel-driven ships with steamships, it should be remembered that there is always a tendency to give the motor-driven ship a slight advantage. These ships are, as a rule, finer than the almost rectangular boxes we are frequently asked to drive by means of steam machinery; and, although I welcome the finer lines as being a step in the right direction, we must bear the fact in mind when we consider the question of propulsive efficiency. I shall be very pleased to give some figures regarding the *Eavestone's* propeller, so that Mr. Knudsen may express his opinion regarding it.*

* PARTICULARS REGARDING PROPELLER, &C., OF M.S. "EAVESTONE."

SHIP—	Length, B.P.	276 feet 0 inches.
	Beam, extreme	40 „ 6 „
	Draught, moulded, loaded	17 „ 9½ „
	Displacement „	4,360 tons.
	Immersed midship area	707·5 square feet.
ENGINES—	I.H.P.	1,075 at sea.
	Revolutions	90 „
PROPELLER—	Diameter	12 feet 3 inches.
	Pitch	10 „ 7½ „
	Expanded area	64 square feet.
	Average speed at sea	8·75 knots.

In the meantime I should like the Secretary to convey to him our very great thanks for the additional information regarding marine propulsion contained in his present paper.

Professor W. Hovgaard (Member): Ladies and Gentlemen, in the absence of the author of the paper, I would like to make a few supplemental remarks, because there are some data relating to the efficiency on sea service of the *Suecia* which were not available at the time the paper was printed. You will notice the *Suecia* has rather fast-running propellers of small diameter, and one might perhaps expect that they would not show up so well in sea service as they had done on the trial trip. The vessel has just completed a voyage to Rio and back again, and Mr. Knudsen gave me a comparison between the results of the voyage of the *Suecia* and of another steamer, the *Axel Johnson*, a steam-propelled vessel of quite similar dimensions, only somewhat finer and longer than the *Suecia*, and of the same power. The trial trip results were the same for both vessels, but on the sea voyage to Rio and back, which was undertaken with only a few days' interval between the ships, the mean result of the speed was rather more favourable for the *Suecia* than for the *Axel Johnson*, showing that the propulsive efficiency seems to have been about the same in the two ships under sea-going conditions. The ships did not, of course, go simultaneously, but with a few days' interval, and since the greater part of the voyage was in the trade winds region, where the conditions of sea and wind are fairly constant, these conditions cannot on the whole have differed much for the two ships. That is not, of course, a scientific comparison, but it is of interest as it throws light on the subject.

The Chairman (Mr. Hall-Brown): I should like to confirm Professor Hovgaard's remarks, and to say that as a matter of fact we do find the sea performance is better maintained with the smaller and more deeply immersed propellers than it is in the ordinary cargo ship with the larger propellers, which are of necessity nearer the surface of the water. I need say nothing more than that the vote of thanks to Mr. Knudsen is, I am sure, a very hearty one.

A CASE FOR ELECTRIC PROPULSION.

By John Reid, Esq. (Member), and H. A. Mavor, Esq.

[Read at the Summer Meetings of the Fifty-fourth Session of the Institution of Naval Architects, June 27, 1913; Mr. E. Hall-Brown, President of the Institution of Engineers and Shipbuilders in Scotland, in the Chair.]

The development of the canal barge or freighter presents a problem of much greater interest and importance than may at first sight appear. No one who travels on the great rivers of Europe, on the Rhine, Danube, or Volga, can fail to observe the immense traffic which they bear, and the remarkable way in which the vessels employed in such traffic have been designed to suit local conditions and requirements. It is so, also, on the great North American inland waterways, the importance of which, as affording the only system of transportation which can compete with the railroad, has been very generally recognised. Perhaps, nowhere in the world is the influence of local conditions on vessel design more recognisable than on the Great Lakes of North America, where the immense quantities of bulk ore, coal, and grain, which have to be handled in a short six months' season, the importance of quick loading and discharge, the shallow nature of the canals and river channels connecting the Lakes, and other local considerations, have led to the evolution of a very remarkable type of vessel, enormous in size, unique in proportions, and unapproachable in the efficiency and economy of its operations. The same influence of local conditions on vessel design is very marked in the Canadian vessels which make use of the St. Lawrence and Welland Canals, by which access by water is maintained between the head of ocean navigation at Montreal and the end of Lake navigation proper at the eastern end of Lake Erie. This canal system comprises the range of canals by which vessels are enabled to pass the currents and rapids of the St. Lawrence River above Montreal, and the Welland Canal, by which the difference in levels between Lake Erie and Lake Ontario, of which the Falls of Niagara are the outcome, is surmounted.

In this series of canals the limiting draught is 14 ft. under normal water conditions, in many of the canal reaches as well as on the sills of the numerous locks. These locks themselves are for the most part of a standard size (Fig. 1, Plate XVIII.), permitting the convenient passage of any vessel not exceeding 250 ft. between perpendiculars, 42 ft. 6 in. beam over plating, and drawing not over 14 ft. in fresh water. Larger vessels have

been locked through, but only with difficulty, and for our present purpose it may be taken that the above dimensions cannot safely be exceeded. When it is considered that a Canadian canal vessel can only operate on the Lakes and canals for six months in the year, owing to prevailing ice conditions, and that for various reasons it is not possible to employ it during the off Lake season in any deep-sea trade, and when it is remembered that the trade in grain, coal, pulp wood, package freight, &c., between Montreal and the Upper Lake ports, and *vice versâ*, is already very large and rapidly increasing, it will be easily understood that not only is it of prime importance to carry the utmost possible deadweight per trip on the limiting draught of 14 ft., but it is equally important to obtain quick loading and discharging and to lose as little time as possible in negotiating the canals and locks of the St. Lawrence and Welland systems.

The last consideration is due to the fact that, as the speed in the canals is limited to four miles an hour, and as there is ever present in the canals the possibility of vexatious delays on account of passing vessels, railway swing bridges, waiting turn at locks, interference from fog or darkness when the main canals cannot be navigated, and so forth, the vessel cannot too quickly escape into open water, where only can continuous navigation at full speed be carried on. Under these conditions it is fair to state that anyone starting out to design or build a canal vessel suitable for the Canadian trade, without reference to the local conditions, would be pretty certain to produce a failure, however successful the vessel might be in open water alone. For example, in attempting to enter a lock which is only 45 ft. wide with a vessel which is 42 ft. 6 in. beam, and under which there may only be an inch of water over the canal bottom, it would be natural to expect that the navigating officer would arrange to approach slowly, pass lines ashore, and gradually warp the vessel in. To follow such a course would probably obviate a good deal of canal damage which these vessels now sustain, but it would inevitably entail such a delay in the locks as would most seriously reduce the vessel's earning capacity. Actually the course followed is for the navigating officer, as he approaches each lock, to line up his vessel as well as wind and current will allow, and to enter at considerable speed, so as to displace the lock water past the vessel's sides as quickly as possible. Full stop a few feet short of the gate ahead is obtained by quick engine reverse, and by holding on with an extra heavy compressor to a snubbing wire made fast to a mooring pin on the quay. When the vessel has risen or descended in the lock, it is equally important to get under way again with the least loss of time, which is usually done by passing a mooring wire forward along the lock wall or canal bank and warping out for a short distance until the propeller can get the vessel up to the desired speed.

It will be at once apparent that under such conditions as outlined above, the form of the propeller, its revolutions and control, must play a most important part in the

efficiency of the vessel as a canaller. Engineers experienced in this line of work look for a screw which will cause this very full-formed vessel to respond immediately to its propeller, and, of course, there goes with that an arrangement of propelling machinery which must respond immediately to the demands of the engineer in charge. A quick-acting throttle and an instantaneous and positive reverse are all important. These would be valueless, however, to the desired end but for a propeller which will enable the vessel to be driven into and out of a lock with the least loss of time. Bear in mind that in entering the lock there is a direct resistance from the water throttled in front of the vessel, and in leaving the lock a corresponding vacuum tendency behind the vessel, and a refusal of the water to flow quickly through the restricted channels between the vessel's sides and the lock walls towards the propeller. Evidently, for successful work under such peculiar conditions, everything depends on the propeller. Note that there is no question here of hull form; that is to all intents and purposes fixed and unalterable. It will be understood that if a Canadian canal vessel has to pass every week through a hundred or more locks, and if the time lost in locking can vary from 70 hours per week for an unsuitable type to 40 for a well-designed and well-handled canaller, it may be well to design rather with a view to overcome the lock and canal delay than from mere considerations of the best speed obtainable in open water, especially as the time which many of these vessels spend in open water is very short.

It has been found, for example, that by the use of a very broad-bladed propeller, with a pitch ratio of 1·1, and at about 80 revolutions per minute, the best results can be obtained in locking operations. The propeller is so designed that it has great driving power ahead and astern in locking operations, quickly decelerating and accelerating the speed in manœuvring into or out of a lock, as the case may be. A type of Lake propeller is shown in Fig. 2 (Plate XVIII.). If one, in imagination, places the vessel shown in Fig. 3 (Plate XIX.) in one of the locks, which it absolutely fills, one will the more easily realise the general helplessness, for efficient results, of any small fine-pitched high-speed propeller, however excellent that type might be in open water work. Furthermore, in lining up to enter a lock, the broad-bladed Canadian propeller can be utilised to throw over the stern of the vessel without occasioning headway, by giving the big, coarse screw a short full-speed impulse with the rudder hard over. This is done as a last resort in entering a lock to prevent a foul entrance, the occurrence of which generally results in a heavy blow on the one bow, a rebound with a blow on the other bow, and finally a severe nip abreast of No. 1 or No. 2 hatch, with the inevitable consequence of serious structural straining, leaky rivets, and damaged cargo.

When the design of the hull (Fig. 3) is carefully considered, it is found that other circumstances, for the most part also arising from local considerations, militate against

any advantage being taken of the fact that these vessels trade in comparatively sheltered waters to reduce the scantlings in an effort to add to the deadweight. The main scantlings, framing, and plating must be fully up to deep-sea requirements, apart altogether from the fact that these vessels, being largely built in the United Kingdom, have to make the passage to Montreal, generally in a two-thirds full load condition, over the stormiest region of the Atlantic. It is essential, in fact, to have even stronger close spaced main framing forward, almost as if providing against ice, to enable the vessel to resist the continual pounding against the lock walls and approaches, to which it is inevitably subjected in this canal trade.

The double bottom must be deep, so as to provide ample water ballast to get the vessel well immersed in ballast trim, otherwise canalling and locking become impossible in a high wind, a great cause of lost time for badly designed vessels. The bottom plating must not be less than $\frac{1}{2}$ in. to resist occasional rubbing in the canal; the tank top plating must be equally heavy to withstand blows from the unloading grabs; the bilges require $\frac{23}{40}$ in. for protection against canal bank rubbing; and the decks must be heavily plated and well supported for deck cargoes of pulp wood and lumber, which may be stacked as high as 14, 16, and even 20 ft. above the deck.

Under these conditions, if an attempt is to be made to increase the deadweight carried on the canal draught of 14 ft.—and owners are perpetually urging such increase—it is found that a limit is soon reached beyond which no reduction in weight of scantlings is possible, and as the form is usually ·83 block co-efficient, or fuller, the deadweight capacity cannot be materially increased by any changes in hull form or structure. In a word, the designer is bound hand and foot as to dimensions, form, and hull weights, and is, therefore, driven to reconsider his propelling machinery in an effort to economise in the weights of machinery and fuel. It is this fact that makes this class of vessel a most natural type with which to experiment with the internal combustion marine engine of the Diesel or other type. The first efforts, however, were devoted to adapting the gas engine and producer, because, while fuel oil is very plentiful and cheap on the Lakes, coal of all kinds, including anthracite, is even more widely available, and at very reasonable rates. Therefore, it was natural to consider the producer-gas proposition in the first place.

In 1908-1909, the marine gas engine had not been reduced to any very practical or reliable form, and the land engines built and running offered very little in the way of experience to guide one in adapting them to marine work. Reliability was doubtful, reversing uncertain, revolutions much too high, and reduction gear at that time by no means to be relied upon. Finally, the attempt to adapt the gas engine was reluctantly abandoned and the experience of other investigators in this field goes to

show that this was an escape from an adventure full of trouble. One thing, however, the writers' investigations did indicate, and that was that no internal combustion engine then extant could be coupled direct to a Lake type of propeller, and that to substitute a suitable type of propeller for the proposed engine, regardless of local requirements in Canadian waters, was to court certain failure under the conditions indicated above. Mechanical reducing gears, though considered, did not offer any great prospect of success, because the reversing gear on the main engines had to be retained, which certainly promised another source of weakness and loss in canalling, possibly a failure to start or to stop under conditions, as regards locks and gates, certain to result sooner or later in serious disaster. At this time electricity seemed to offer possibly the best prospect of a successful solution to this problem. Being satisfied that the single Lake type of propeller and the facilities for manœuvring the same under steam would have to be retained if failure was to be avoided, it remained to find some third feature through which the desired large slow-running propeller could be reconciled with the comparatively light high-speed internal combustion engine of the Diesel or other type. It must be admitted that the introduction of electricity for this purpose, while committing one to a certain loss in transmission, gave assurance of some incidental advantages of great value, which had not originally been anticipated.

No internal combustion engine, Diesel or other, known to the writers, is an absolute substitute so far as simplicity of operation and reliability of handling are concerned for the ordinary up-to-date triple-expansion steam engine, even in deep-sea go-ahead work, much less in manœuvring in narrow waters within harbour limits or in rivers or canals. Whatever success may have been obtained by the recently developed two-cycle, slow-speed, marine type of Diesel engine (and there is here no suggestion that success has been other than most encouraging), it still seems that the line of development followed is not such as seems likely to lead to the evolution of a propelling engine which an owner will be as ready to have in his vessel as he would an ordinary steam outfit. Attention has been paid to obtaining a specious resemblance to the marine steam engine in an effort to make the Diesel engine exactly suited to take the steam engine's place direct on the propeller shaft for the sake of a simplicity which is more apparent than real. To get the low revolutions most desirable for propulsion in slow-speed cargo vessels, important advantages in the normal Diesel engine, such as high speed, low weight, and moderate over-all dimensions, have been sacrificed, while the necessity for fitting reversing gear on the engine itself has added a most unfortunate complication necessitating the lavish use of compressed air in manœuvring, an addition of a particularly troublesome and expensive nature. Furthermore, experience has already shown that the use of large single-screw Diesel engines in propulsion is attended with grave risk of the breakdown of the whole propelling mechanism of the vessel.

It is idle to deny that there is inherent in the very operating principle of the Diesel engine an element of delicacy and unreliability very little in keeping with the rough work of marine propulsion from which the modern steam engine emerges with credit. The fuel economy which has been the one real outstanding claim for the adoption of the Diesel engine is, in fact, dependent on an operation of great nicety involving the use and maintenance of very delicate mechanism. A governor action controlling this mechanism is set in motion by the racing of the engine in a seaway, and the cycle of the cylinder operations, so far from continuing uniform as is essential for good working, are periodically interfered with, probably under impulses given too early or too late, with inevitable trouble. It must be considered also that no engine of the Diesel type, with its numerous spring-loaded cam-driven valves and other fine-set gear, can take kindly to the vibration set up by the action of a vessel in a seaway. The reference here is particularly to a vessel engined astern, but the same is true of any position of the machinery in direct drive. Finally, any adjustments necessary on the Diesel engine, the need for which may not be at once apparent—in fact, which may not become apparent at all until serious trouble has developed—must be at once attended to, to avoid trouble, whereas in a steam engine they may without serious detriment be put off to a suitable opportunity.

Enough has been said to indicate some of the points which experience has already developed as requiring consideration in the present Diesel type of marine engine before its reliability can be confidently placed in the same category with that of the triple-expansion steam engine. After all, it is the average shipowner's wish to have a ship which, while attaining all reasonable efficiency and economy in its operations, shall be, first and foremost, trustworthy, and the propulsion of which will be carried on as surely as the limitations of even the best engineering mechanism will permit. It is hopeless to introduce new methods of marine propulsion which do not guarantee all the security and certainty of the mechanism which they displace. Therefore, it is suggested that it is unfortunate that so much effort has been devoted to attempting to force the Diesel engine into conditions of service for which it appears inherently unfit, without saying that in certain classes of vessels where the conditions are special there may not be a perfectly legitimate field for the type criticised, viz., the direct-drive, slow-speed, two-cycle open marine engine. Such scope is most assuredly not found in the full-bodied cargo boat, still less in the Canadian Canal vessel under the peculiar conditions and limitations above described. It is unthinkable that, with its present form of hull, which cannot be varied, and the still more definitely fixed type of propeller, the present triple-expansion engine can be displaced by any directly connected Diesel engine. Eighty revolutions per minute, which is common practice with the steam engine, is almost

hopeless for the Diesel engine, and however successfully the reversing gear may operate in the latter, it is little likely to be improved by the hundreds of rapidly alternating go-ahead and go-astern motions required in negotiating such a series of locks as those in the Welland Canal.

Broadly speaking, the introduction of electricity in this connection puts the Diesel engine back into work for which it is eminently suited, under conditions which give it every chance to maintain efficiency and reliability. It is as far as possible removed from the uncertain action of the propeller, and the uneven loading and vibration incidental to a direct propeller drive. In Canadian Canal work the power required for propulsion varies between very wide limits. In the canals themselves, four miles an hour being the limiting speed, 150 I.H.P. or less is sufficient, though full speed must be always available for emergency, especially in backing. In the open lake in fair weather 500 I.H.P. is sufficient power to maintain fair speed, while 750 I.H.P. is necessary in contending with the St. Lawrence currents. In some designs the writers, therefore, proposed three Diesel generating units of 250 I.H.P., each of which can be switched on to the propelling motor as required. Such an arrangement is shown in Fig. 4 (Plate XX.). In the *Tynemount*, which is now building, to demonstrate the Diesel electric arrangement of propelling machinery, a two-unit subdivision is arranged for, each unit furnishing 300 B.H.P. Fig. 5 shows approximately the general layout of the installation, and attention is directed in this illustration, and also in Fig. 4, to the ease with which the generating units can be located in any desired position, in a separate engine-room remote from the propeller motor, or on an upper flat, or wherever may be considered convenient.

The electric system adopted is that identified with the name of the second writer, involving the use of alternating current. Investigation and actual tests have amply shown the suitability of this system for marine propulsion.

This system has for its special object the simplification of the electric equipment. Many applications of electric power have been made to marine propulsion, but hitherto, with the exception of the *Electric Arc*, propelled on this system and built in 1911, continuous current only has been used, the reason being that regulation of speed and control is easier for continuous current under ordinary conditions than for alternating current. The disadvantages attached to the use of alternating current in respect of regulation are associated with the greater number of conductors and particularly with the property of alternating current motors, that the speed of the motor bears a fixed ratio to the speed of the generator, and that any departure from this speed is associated normally with loss of efficiency and with more or less complicated devices for changing the frequency of the current alternations.

In the system under notice these difficulties are overcome by the use of more than one frequency applied to each individual motor. The currents of different frequency

are carried in independent mutually non-inductive circuits, the magnetic systems being entirely independent, but operating upon a common rotor, so that their mechanical effects can be superposed and the power transmitted in the separate circuits combined to produce the required mechanical effect. The required currents may be obtained from one or more generators.

In the vessel now under description (the *Tynemount*) the equipment consists of two three-phase generators driven by Diesel engines running at 400 revolutions per minute. The electrical output from each set is 235 kilovolt ampères at 500 volts alternating. The generators have six and eight poles respectively, giving frequencies of 20 and 26·6 per second. The exciting current is obtained from direct connected continuous current machines on the same shafts as the alternators. The normal exciting current is 30 ampères at 100 volts. A single three-phase motor is coupled direct to the propeller shaft, which is of the ordinary type with marine thrust block. The motor develops 500 S.H.P. The rotor or moving part is of the simple squirrel cage type, without any electrical or mechanical connection other than its rigid attachment to the propeller shaft. The stationary part of the motor has two separate non-inductive windings for 30 and 40 poles respectively. When each of these two windings is connected to the appropriate generator, the speed due to each is 78 revolutions per minute. By changing the connection on both windings, the direction of rotation is reversed, and by connecting the 40-pole winding of the motor to the 6-pole generator the speed in either direction drops to 58 revolutions per minute, or about three-quarters of the full revolutions. If full speed be not required, one generator may be stopped and the other left running at full revolutions under governor control and at full economy, because the power required to drive the ship at three-quarters speed is about half of that required to drive it at full speed. If either of the generators be left attached to its own winding and the other generator shut down, the ship is propelled by either engine at a little over half speed, the speed of the ship falling with the speed of rotation of the engine, until an automatic adjustment of power and speed is reached at about half-speed.

The control gear consists of an apparatus for changing the connections of the generator and motor windings respectively. There are five positions on the switch, corresponding to the ordinary positions on the engine-room telegraph. They are "Full Speed Ahead," "Half Speed Ahead," "Stop," "Half Speed Astern," and "Full Speed Astern." Each position of the controller is definitely fixed by means of cams and rollers, so that stopping at intermediate positions is prevented. The controlling gear provides for the interruption of the excitation of the generators while the switch is being moved from one step to another, the exciting switch and the main switch being interlocked so that the switching over operation is accomplished while the electric

circuits are "dead," thus avoiding injurious sparking. Should it be found convenient, a very simple arrangement could be made whereby the control could be operated from the bridge of the vessel, and the engineer's attention confined to the keeping of his engines lubricated and supplied with fuel to run at constant speed.

It will be seen that this method of control entirely dispenses with the stopping and starting of the Diesel engines for manœuvring, an operation which, in itself somewhat difficult, becomes impossible if a liberal supply of compressed air be not available. To keep up a supply of compressed air for manœuvring the vessels in the locks and channels of a canal involves the upkeep of a very expensive and inefficient air-compressing plant, and the dispensing with this auxiliary is a very important feature of the system. It is also advantageous to have two units, each capable of driving the ship, so that in the event of any interruption to the running of either the vessel is still under control.

The functions performed by the electric equipment may be recapitulated :—

(1) It adapts the speed of the engine to the speed of the propeller.

(2) It combines the power of separate engines and applies the whole to a single propeller, with perfect freedom to use either or both power units.

(3) It provides a simple and easy reversal of the propeller, while leaving the engines running in one direction at constant speed.

(4) It also provides ready means of distant control should this be required.

It will be seen that the use of mechanical gearing could perform the first and the first only of these functions, and for this reason it is anticipated that, compared with a mechanically geared or directly connected Diesel engine equipment, the electrical equipment will offer very material advantages in the operation of the vessel in the special circumstances under which it is placed.

The writers have endeavoured to show how important, and, indeed, even indispensable for reliability and efficiency, the use of electric transmission may be in the propulsion of so relatively simple a type as this Canadian barge where it is decided to adopt the Diesel engine. This should help to dispose of the too prevalent idea that the natural scope for electric propulsion is in warships or other highly specialised vessels in which electricity might provide means for adjusting economically the power generated to the very conflicting power requirements, say, under peace and war conditions. Undoubtedly a battleship or battle cruiser does offer the very best scope for the electric drive, but only because that class of vessel presents on a large scale the same problems in propulsion which are found in almost all other vessels, viz., how

to adjust efficiently and economically the power available to the power required at any given time, not necessarily always from the point of view of propulsion alone.

From the analysis of the voyage logs of an ordinary tramp steamer, the second writer has shown (see *Electrician*, June 10, 1910) how important towards efficiency of propulsion might be the introduction of electric transmission, with the added economy of a modern high-speed reaction turbine as the power generator in place of the usual triple-expansion engine, and, of course, in this case such possibilities of economy are confirmed by the results since obtained with mechanical gearing introduced for the same purpose. An advantage, however, lies with the electrical arrangement, in that it permits a much wider range and variation of revolutions between the propeller and the turbine, permitting also a more advantageous subdivision of the generating plant into units of different sizes, and finally eliminating the necessity for reverse in the turbine and simplifying the control.

In the United States, Mr. Emmett, of the General Electric Company, has also shown the suitability of electric propulsion for a deep-sea collier of great size and 7,000 H.P., for which the equipment has already been completed and tested with the most brilliant results. From such a vessel to a large passenger liner such as the *Celtic* is a mere step, and in such vessels also the possibilities of economy and all-round efficiency are most marked.

In such matters it is much too common to pronounce judgment for or against without adequate analysis of the problems which are as numerous and varied as the vessels which come up for consideration. In this connection, trial-trip data, however valuable for comparative purposes, cannot be used as a guide, because what we are concerned with is maximum efficiency in working conditions; not propelling efficiency alone, but the innumerable phases of the working of a ship, of which propulsion is, perhaps, the most important, all of which have a bearing on the general efficiency of its operation. Experience in developing a comparatively simple type, such as this Canadian Canal vessel, shows that it is frequently quite impossible to convey to the builder any adequate conception of how such a vessel is handled, and has to be handled, to get maximum efficiency out of it in the short six months' season. What Lord Kelvin used to call the "bias of preconceived notions" is apt to prevent a detached view on the part of a builder, especially when a decided departure from established practice and precedent is called for. For example, the idea that special scantling provisions forward should be fitted to any type of vessel to enable it to "butt" stone walls and quays with impunity seems to many builders very unreasonable, though the same builders might not hesitate to fit special bow stiffening against ice.

It is for this same reason that one is not surprised to meet with considerable scepticism in regard to the scope for electricity in marine propulsion. To the majority of sea-going engineers the idea of a transmission feature between the driving engine and the propeller is simply anathema. To prevent such an introduction an infinity of pains will be devoted to experimenting with high-speed propellers and low-speed turbines and oil engines, with an all-round lowering of total efficiency and an increase in complication actually in excess of that which it is desired to avoid. There are not 10 per cent. of the merchant vessels now afloat which would not be most efficiently propelled by screws designed to turn at revolutions not exceeding 80 per minute, and for real progress towards maximum efficiency in marine propulsion one should at all costs retain the simplest form of propeller, the efficiency of which permits of no dispute, and from that basis arrive through the various methods of transmission now available at an arrangement of machinery best suited to each individual case. At the start such a scheme of powering a vessel would work a hardship on the builders of standard marine machinery; but it should be understood that, in advocating the use of transmission gearing, one does so to enable the newer forms of power generators to be utilised.

Nothing is more likely to retard the introduction of the internal combustion engine in marine work than the mistaken attempt to treat it as a perfect substitute for the triple-expansion engine which it is intended to displace. It is at this point that electricity acts as a safeguard in a way that no other system of transmission can approach.

There are, of course, problems of propulsion, the analysis of which shows that approximately equal results may be obtained whichever form of transmission gearing, electric, hydraulic, or mechanical, is adopted. In such cases the decision will no doubt favour that system which offers the greatest reliability and simplicity of operation. The success which has attended the introduction of the geared turbine and the hydraulic transformer is of good augury for similar success with electric transmission. But there is a class of problem in marine propulsion for which the only possible solution is through electrical means, viz., vessels in which propulsion is only one, and sometimes not the most important one, among many functions calling for power, which it is evident would be most efficiently provided from a central generating station. Among such vessels one might class the larger sizes of warships, with their multitudinous requirements in power for the driving of auxiliary engines as well as propelling machinery, but for purposes of illustration it is probably better to take a simpler case, such as a dredger, say, of the self-propelled suction type. Attention was first called to the suitability of electric drive for such vessels by Professor Biles and Mr. H. A. Mavor, and Mr. John Reid has recently completed a design for a large vessel

of this class which clearly indicates the very important advantages which may be obtained by the introduction of a central generating station, furnishing power as required to the propellers, the dredge pumps, or the numerous auxiliary machines.

Fig. 6 (Plate XIX.) shows the general layout of the engine-room, in which there is a single generating turbine plant with electric motors on the screw shafts and also on the dredger pumps. In a similar steam-driven job there are four separate triple-expansion engines, two for propulsion, each of 800 I.H.P., and two for the pumps, each of 500 I.H.P., or a total of 2,600 I.H.P. When it is considered that the introduction of the electric system allows the total horse-power generated to fall back to 2,000 as a maximum without in any way prejudicing the operations of the dredger, it is at once apparent that here we have a first-class incentive to adopt an electric arrangement. A further incentive of great importance in such work is the ability which the use of electricity confers to place the complete control of the dredger in the hands of one man, the dredger master. Here, again, we have evidence of the importance of studying such a problem rather from the side of the dredger operator than from that of the dredger builder. No real progress can be made until this is done. By a simple system of switches scarcely more complicated than ordinary bridge telegraphs, one man can handle the whole machine directly with the very greatest advantage to the efficiency of the dredging operations.

Such are some of the most likely cases in which electricity seems to offer a means of solving certain difficulties in the power question, which are, after all, only components of a problem which in one form or another has long confronted the marine engineer, viz., how to generate the power required in a given vessel independently of the means used for propulsion, enabling the source of power for all purposes to be concentrated in a central station with a simple economical and easily controlled system of distribution, by which the power generated for any particular purpose may be exactly related to its requirements.

APPENDIX.

Average Log for Package Freighter of "Canal" Type. To Show Fluctuation of Power Requirements.

Montreal to Fort William and Return.

Leaves Montreal with about 700 tons freight; draughts, say, forward, 10 ft. 1 in.; aft, 7 ft. 4 in. From Harbour of Montreal enters Lachine Canal, 8½ miles long, 5 locks, speed limited to 4 miles per

hour. Time taken, 4 hours. Crosses Lake St. Louis, 15 miles, no current. Time taken at 10 miles per hour, 1½ hours.

Soulanges Canal.—14 miles, 4 lift locks, 1 guard lock. Time taken, 6 hours.

Crosses Lake St. Francis.—32 miles, currents 1 to 3 miles per hour. Time taken, 4 hours.

Cornwall Canal.—11 miles, 6 locks. Time taken, 5 hours.

Williamsburg Canals.—A series of canals alternating with stretches of river, with 5 to 6 mile currents. 12 miles. Time, 5 hours.

To Prescott in river.—30 miles; 5 mile current; 4½ hours. Distance from Montreal to Prescott, about 125 miles, usually done in 26 hours, depending on weather and time of leaving Montreal. Allow 30 hours, in which at least 10 hours vessel is stopped altogether, locking and tied up.

Prescott to Toronto.—230 miles, full speed; about 21 hours. Stop at Toronto for cargo. 25 to 35 hours, say 30 hours. Leaves Toronto with 1,200 tons cargo; draughts, forward, 11 ft. 9 in.; aft, 9 ft. 10 in.

Toronto to Hamilton.—40 miles; 4 hours. Stop at Hamilton for cargo 25 hours. Leaves Hamilton with 1,400 tons cargo; draughts, forward, 12 ft. 7 in.; aft, 10 ft. 8 in.

Hamilton to Port Dalhousie.—30 miles; 3 hours.

Welland Canal.—26¾ miles, numerous locks and swing bridges; 16 to 24 hours, depending on wind and number of boats coming down. Wind is a bad drawback, owing to exposed nature of Canal. (Vessel is driven over against bank); say 20 hours.

Port Colborne to Courtright on St. Claire River.—300 miles; 28 hours. Stops at Sandwich for coal and picks up freight at small ports. Add 13 hours stopped.

Courtright to Sarnia.—10 miles; 1 hour.

Sarnia to Soo Canal.—270 miles; 26 hours. Allow about 5 hours average delay at Soo taking cargo, waiting for locks, &c.

Soo to Fort William.—260 miles; 26 hours full speed.

Total distance, 1,300 miles; usual time, 9 to 10 days. Loading grain at Fort William, 3 days. Return journey to Montreal, 7 days. Unloading Montreal and waiting turn, 2 days. Loading package freight, 2 days. Total for round trip, 24 days. Actual average is rather longer, 25 to 27 days, owing to delays, Sundays, &c. Fuel consumption, 10 tons per average day right away.

Report on Trip on S.S. "Keystorm." Montreal to Oswego.

Beginning July 22. *Ending July* 25, 1910.

Although our engineer did not join the *Keystorm* until she reached Lachine, we have the approximate time at the first locks between Montreal and Lachine. Unfortunately during the trip there were unavoidable delays caused by breaking of Lock 18 in the Cornwall Canal. This in itself not only delayed us, but tied up a number of boats which all got away together, thus causing further delay in waiting for locks and passing these boats in the Canal.

The low power cards were taken in the Soulanges Canal. The canal here is 11 miles long, without a lock, and has power stations at the lower end, which makes a current of at least 1½ miles per hour. The speed of the ship was accurately measured by taking the time between posts marked and fixed in the canal bank ½ mile apart. This distance the boat made in 7 minutes and 45 seconds at fifty revolutions, which is a speed of 3·87 miles an hour, or practically 4 miles an hour. The draught was 11 ft. 7 in. aft and 13 ft. 2 in. forward. The indicated horse-power as shown by cards was 218.

There are canals without currents but they are short, and it would be impossible with only one indicator to get the cards in time. In any case it is a difficult matter, as they are constantly changing the speed of the engine. While the boat was in the currents the engines were turning 82 revolutions per minute, 177 lbs. of steam, 48 first receiver, 9 lbs. second receiver, 25½ in. vacuum.

Cards were taken on the Lake at the same pressure and revolutions as when running in the currents. The total power developed was 752 I.H.P.; speed about 10·25 miles per hour. The coal was accurately measured for 6 hours, and, allowing 44 cubic ft. to the ton, amounted to 15·6 tons per 24 hours, or about 1·9 lbs. of coal per indicated horse-power per hour. The coal was of inferior quality, with a high percentage of ash.

The speed of the currents varies, therefore it is hard to get any accurate information regarding their speed by taking the speed of the ship.

On this trip it was noticed that the boilers were being operated under natural draught; while running slow speed and in the canals. This is not good practice and very uneconomical, no benefit being derived from the heating arrangement in the uptake. When working very slowly, the fan if making too much draught can be stopped, but the ash pit dampers should always be shut, the necessary air being taken through the fan and uptake. Any quantity of ashes should never be allowed in the ash pit, otherwise the fire bars will get red hot and collapse.

It was also noticed that the injector was used excessively in canals. This is a very inefficient instrument for feeding boilers, especially when a feed pump is in operation and doing the same work. An injector should only be used for feeding salt water or to fill a boiler when all feed pumps are cold, if economy is to be considered, as most of the steam is used to give velocity to the water. Some of these things are common practice, but are not right. The engineer has been instructed on these different points.

In the following report the locks will be numbered from Montreal, counting only the locks the boat passes through. The ordinary numbers include some old canals, and are rather misleading.

Lachine Canal.

Total length, 8½ *miles*. *Total elevation of lockages*, 45 *ft*.

Lock 1.—In and the gates shut 8.35 a.m. Clear of the gates at 8.55 a.m.

Lock 2.—This lock is straight ahead, about 100 yards distant. In and the gates closed at 9.23. Clear of the gates at 9.42 a.m.

Lock 3 (Gabriel).—This lock is about a mile distant. The canal is comparatively wide in places, at other parts it is rather narrow, and crooked. There is one bridge immediately on leaving Lock 2. There

are also elevators, coal and freight wharves along both sides where boats and barges are tied up. In and gates shut at 10.4. Clear of gates at 10.20.

Lock 4 (Côte St. Paul).—This lock is about 1½ miles distant. The canal is narrow all the way, but comparatively straight; has one bridge and dredges working to repair canal bank. In and gates closed at 11.15. Clear of gates 11.30.

Lock 5 (Lachine Lock).—This lock is about 6 miles distant; has 2 bridges and is comparatively straight, but from these canals at different points there is water taken for power purposes, which, besides making an undesirable current, causes eddies around, and in the immediate vicinity of the outlet strong suction, which sometimes draws a ship close to the outlet, when the power has to be shut off before she can be taken away. In passing these a ship must keep well to the opposite bank. In and gates closed at 1 p.m. Clear of the gates 1.15.

Lake St. Louis.

On leaving this lock there is a canal of about 1½ miles in length. Cleared this at 1.40, after which a boat can usually run at full speed on the Lake, although it is only a channel marked by range lights and buoys. There is a current in this channel estimated at a mile an hour, but during the last 4 miles the current is stronger, varying from 1 to a maximum of 3½ miles. The distance is 17 miles.

Slowed down at 3.46 and tied up at the entrance of the Soulanges Canal at 4 p.m., making a total time between locks of 2 hours 45 minutes. The boat was running at half speed for about one-third the distance. This was on account of one of the lightships being out of its position.

Soulanges Canal.

Length, 14 *miles. Total elevation of lockages*, 84 *ft.*

Lock 6.—This lock is straight ahead about 200 yards. In lock and gates shut at 4.48. (It takes on an average 8 minutes for a loaded boat to enter lock.) Lock filled, gates opened and engine started at 5 p.m.

Lock 7.—This lock is about 300 yards straight ahead; no current or obstruction of any kind. In lock and gates shut at 5.20. Lock filled, gates opened and engines started at 5.30.

Lock 8.—This lock is about 2 miles distant; has one bridge, but no other obstruction. In and gates shut at 6.16. Out of this lock and through guard lock, about 300 yards distant, at 6.35.

Lock 9.—After passing through Lock 8, tied up until 7.30 to allow the *Rapids King* and R. & O. passenger boat to pass. From this lock to Lake St. Francis or Côteau Landing, a distance of 11 miles, is where the cards were taken. It has at the lowest estimate a current of 1½ miles. There are two bridges and some construction plants fixing the canal bank. In and gates shut at 10.23. Lock filled, gates opened and engine started at 10.35.

Lake St. Francis.

This lake is 28 miles long; has very little current except at a distance of about 4 miles from the west end, where it varies from 1 to 4 miles per hour. Arrived at Cornwall 2 a.m., Saturday, July

23. Here it was found that the third lock from here was broken (called 18), therefore tied up for the night, causing a delay of about 14 hours.

Cornwall Canal.

Total length, 11 miles. *Total elevation of lockages*, 48 ft.

Lock 10.—Under way again at 4.10 p.m. In lock and gates closed at 4.30. Some trouble in getting gates opened. Lock filled, gates opened and engines started at 5.25.

Lock 11.—About 250 yards straight ahead. In lock and gates closed at 5.35. Lock filled, gates opened and engines started at 5.47. After leaving this lock the ship tied up again, as the next lock was broken and it was not then repaired. Delay, 5 hours.

Lock 12.—About 7½ miles distant. Two bridges, slight bend about ½ mile after leaving lock. Left Cornwall 11.15, arrived broken lock at 12 midnight. In broken lock and gates shut at 3 a.m. July 24 (delay, 3 hours).

Lock 13.—About 1¼ miles distant, ¾ of a mile straight before entering lock. In and gates closed at 4.34 a.m. Lock filled, gates opened and engines started at 4.50.

Lock 14.—About 1¼ miles shallow water and crooked channel. In and gates shut at 5.30. Lock filled, gates opened and engines started at 5.45. Passed through guard lock at 5.55.

Lock 15 (Dickinson Landing).—About 7½ miles. Crooked, but plenty of water. In fact, it is a branch of the river made into a canal by means of dams. One bridge. Arrived head lock. In and gates shut at 7.40. (Lift only 3 ft.) Lock filled, gates opened and engines started at 7.50.

Lock 16 (Farran's Point).—One lock, 800 ft.; lift, 3½ ft. After leaving Dickinson's Landing there is a breakwater about ½ mile, with a light on the outer end. Passed this light 8 a.m. From here to Farran's Point, a distance of 8 miles, a boat is in a current of about a maximum of 4½ miles. Engines slow at 8.35. This is the worst landing to make in the whole river; in fact, a captain would not make it at night with a loaded boat (14 ft.). A swift current flows past the entrance of the dock, causing a miniature tide in the lock, and eddies between the current and the bank. There is a rise and fall at the lock of about 8 in. in every 10 minutes. In lock and made fast at 9 a.m. Waited for other two boats. Lock filled, gates opened and engines started at 9.40.

Lock 17 (Morrisburg).—Two locks, 11½ ft. lift. After leaving Farran's Point there is a canal about one mile long. Clear of this at 10.10 a.m., and into the river and currents. Passed Weaver's Point at 10.40, the first bad point. Current estimated at 8½ miles. Passed the Churches, the second bad point, at 11.20. Here a boat crosses from the south to the north shore, and, of course, loses some ground in doing so. From here to Morrisburg, a distance of about 3 miles, there is a current of between 6 and 8 miles. Total distance from Farran's Point to Morrisburg 12 miles. Dorin's Point, just below Morrisburg, is about the swiftest current in this run. Slow 11.50. Tied up waiting for lock gate at 12.05. In lock, gates shut at 12.23. Lock filled, gates opened and engines started at 12.37. A boat having a speed of 9 miles per hour could leave the head of Cornwall Canal, going around Farran's Point up the Snye and land at Morrisburg. There is no time gained or lost in this, but it avoids a difficult landing at Farran's Point.

Lock 18.—Head lock; lift about 1 ft. About 4 miles distant; crooked, with slight current. In lock and gates shut 2 p.m. Lock filled, gates opened and engines started at 2.10.

Lock 19 (Iroquois).—Two locks, 15 ft. elevation; 1,800 ft. long. From this lock it took 5 minutes to clear the canal. (2.15 p.m.) From here to Iroquois, a distance of 7 miles, is in the river and currents. Passed Pine Tree Point at 2.45, which is as swift a current as there is in the river; the boat barely moved in the worst part, but it is only about 300 yards long. Slow 3.05, waited for another boat. In and gates shut at 3.30. Lock filled, gates opened and engines started at 3.55.

Lock 20.—About 7 miles distant; crooked at the eastern end and rocky bottom. Very few iron boats loaded venture through here at night. Two bridges. About 2 miles east of Cardinal there is a rough stone wall on either side. These are close together as no boats pass down, all are up bound. In Galops Lock, gate shut 5.50. Lock filled, gates opened and engines started at 6.10. Soft bottom in this lock. A boat is not allowed to work her propeller until she is about half out of the lock. This is the last lock. After leaving this lock there is a canal about ½ mile long, then the open river with a current of about 3½ miles for about a mile and through another canal called the "North Channel," about 1½ miles. Passed the western end of this 6.43 p.m. Prescott, 5 miles, at 7.20. From here along for about 20 miles is the Thousand Isles. Arrived at Oswego at 7 a.m., July 25. The distance from Prescott to Oswego is about 125 miles.

DISCUSSION.

Mr. Max Wurl (Member): Ladies and Gentlemen, Mr. Reid and Mr. Mavor have put before us a case where electric propulsion would have certain advantages compared with other methods of propulsion, and no doubt the arguments they bring forward have some force behind them; but I think, in comparing this method of propulsion with the ordinary direct drive by means of Diesel engines, they rather over-estimate the difficulties. The authors have especially pointed out difficulties in the case of propellers, and I would like to ask one question with regard to the statement on page 158 where they say: "It has been found, for example, that by the use of a very broad-bladed propeller, with a pitch ratio of 1·1, and at about 80 revolutions per minute, the best results can be obtained in locking operations." This is a very definite statement, and I think it would be interesting to the Institution to know whether any definite comparative trials have been made by which such figures as these have been obtained.

They also speak about the helplessness of small fine-pitched high-speed propellers, and it is stated on page 161 that the Diesel engines are inherently unfit for this particular trade. If this were the case I think it would be rather a bad look-out for the Diesel-engined ships that have gone to the Lakes. There is one ship which is particularly interesting in connection with the *Tynemount*, viz., the *Calgary*, which is practically of the same size and power. She went out last year and has made several voyages on the Lakes. She has two propellers of very fine pitch running at 250 revolutions a minute, and yet the results are evidently as good as those obtained with steamships. I will only mention the locking through the Welland Canal; it is stated in the Appendix that the time taken for going through this Canal is generally 16 to 24 hours. On reading this paper I looked up the log abstract for the *Calgary*,

and found this year she has done five passages through the Welland Canal, and the times taken were 18, 17, 16½, 17½, and 18 hours, which compare very well with the 16 to 24 hours mentioned for the steam vessels. It is not impossible, however, that electrical propulsion may be able to improve on these figures, and it will be most interesting to compare the results of the *Tynemount* later on.

Mr. H. A. Mavor: Mr. Chairman and Gentlemen, with regard to Mr. Wurl's remarks I would like to say one word. One cannot judge of conditions of working from the log alone, and therefore I had a look at the question on the spot in the engine room of the vessels making these Canal passages. There is no doubt in my mind that a propeller running at 250 revolutions which has to be reversed direct by the engines provides a heavy task for the engineers, and they are not slow in saying so. The slow-running propeller at 80 revolutions is the one which has proved most satisfactory in practice. That I may say from my own knowledge, although the statement in the paper really depends for its authority entirely upon Mr. Reid.

The Chairman: Gentlemen, I am sure it is in accord with your wishes that we should render a very hearty vote of thanks to Mr. Reid and Mr. Mavor for this paper, and for what we are hoping still to receive in the way of additional information.

Written Communication.

Mr. John Reid (Member): Mr. Wurl has referred to the *Calgary* now operating on the St. Lawrence canals and the Lakes with twin-screw reversible Diesel engines in direct drive. I am familiar with the operation of the vessel, which has proved fairly successful in ordinary conditions of working. It has certainly not been found, however, that the *Calgary* arrangement of machinery obviates the difficulties that are encountered in the Canadian canal trade to which I have referred. It has been found, for example, that this vessel, in the loaded condition, cannot stem the rapid currents of the St. Lawrence, which steam vessels of precisely the same dimensions and power negotiate easily. That is a proof, if any were needed, that the small twin-screw propellers fitted in this vessel and turning at 200 revolutions per minute and upwards are considerably less efficient than the single Lake type of propeller with steam-engine drive. It is our belief, which actual trials in Canada ought to justify, that the *Tynemount* will prove as handy and efficient, under these peculiar local conditions, as the steamers of the same dimensions and power. That is the only reason for introducing the electric equipment.

CONCLUDING PROCEEDINGS.

Mr. W. H. Whiting (Member of Council): Gentlemen, I have been asked to discharge a very pleasing duty this morning at the conclusion of our proceedings, and that is, to propose a vote of thanks for all the kindness which we have received in every shape and form during this week. This is a pleasant but at the same time a difficult duty, for it is a very easy thing to say more than you intend in matters of criticism or argument, but it is not easy to say more than one intends, or more than one ought to say, when speaking on behalf of the recipients of such kindness as we have enjoyed this week. From the very first stages of the proceedings, when we were received so cordially by Lord Inverclyde, by Mr. and Mrs. Hall-Brown, and the members of the Reception Committee, we have enjoyed uninterrupted kindness, which has been shown to us both in an official and an unofficial capacity. We have received from our friends every token of welcome that we could desire, and I am sure that many of us will long look back to these meetings with the utmost pleasure. The main object of these meetings has been admirably discharged: we have had most excellent papers, some of which will, I think, long be quoted and used by those who peruse our Transactions. We have, moreover, been fortunate in having enjoyed while in Scotland the best of weather, and I am sure if there was anything that could add further to our happiness this week it has been that circumstance.

I wish to propose, on behalf of the Institution of Naval Architects, in the most sincere and unqualified terms, a hearty vote of thanks to the Institution of Engineers and Shipbuilders in Scotland for all the facilities which they have given us, and for all the kindness which, from the President downwards, they have so graciously bestowed upon us. I ask you, gentlemen, in passing this vote by acclamation, not to forget the Secretary of the Reception Committee, Mr. Parker, whose efforts have so greatly contributed to the success of our visit.

Mr. S. J. P. Thearle, D.Sc. (Vice-President): Mr. Chairman and Gentlemen, I quite concur in all that has been said by my friend Mr. Whiting in giving expression to the grateful appreciation which we, the Members of the Institution of Naval Architects, feel in regard to the reception we have met with from the Members of the Institution of Engineers and Shipbuilders in Scotland. We feel it very deeply, and, as Mr. Whiting has said, it is difficult to give full expression to the feelings of one's heart when the heart is overflowing. I do not think I could add a word with advantage to what Mr. Whiting has said, in seconding this resolution which he has proposed. I would only say that, while thanking you for the cordiality and the thoroughness of the efforts you have made to entertain us and instruct us, I confess that I envy you this magnificent building and the rooms it contains. I hope that we in London may some day have the wisdom to imitate you, at least in some degree. I join very heartily in seconding the vote of thanks which Mr. Whiting has so well proposed.

The Chairman (Mr. Hall-Brown): Mr. Whiting, Dr. Thearle, and Gentlemen, I desire to thank you on behalf of the Institution of Engineers and Shipbuilders in Scotland for the kind things you have been pleased to say regarding the arrangements for your Summer Meeting. That these arrangements have met with your approval, and have added to your pleasure and to the success of your meetings, is a source of gratification to our Institution and to the Reception Committee. We are happy in knowing that we have contributed to your happiness and are amply rewarded by your kind appreciation. I am very glad indeed that in this motion you have referred to the work of our Secretary, Mr. Parker.

I assure you that those who come into close contact with Mr. Parker esteem him most highly, and the success of these meetings has lain very closely to his heart. I would also like to say, that in all the negotiations and arrangements connected with these meetings we have had invaluable assistance, and in many instances the happiest advice, from your esteemed Secretary, Mr. Dana, whom I wish to thank most heartily.

Mr. R. W. Dana (Secretary): I should like to thank you, Mr. President, for the kind remarks you have just made, and to entirely endorse what Mr. Whiting and Dr. Thearle have said with regard to our splendid reception in Glasgow, and also to refer to Mr. Parker's untiring and most successful efforts, which have contributed largely to make this week such an extremely pleasant one for all concerned, and one which will live long in the memories of those privileged to take part in these meetings.

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of Steamers."

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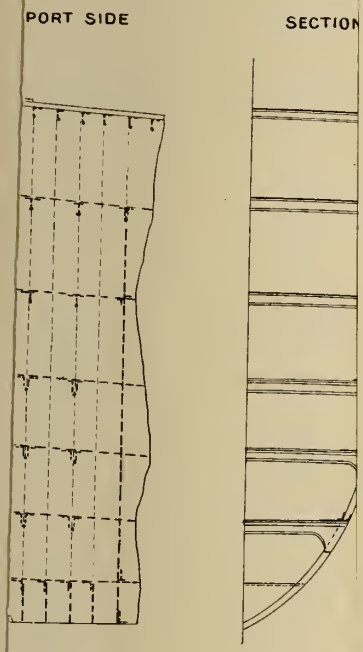

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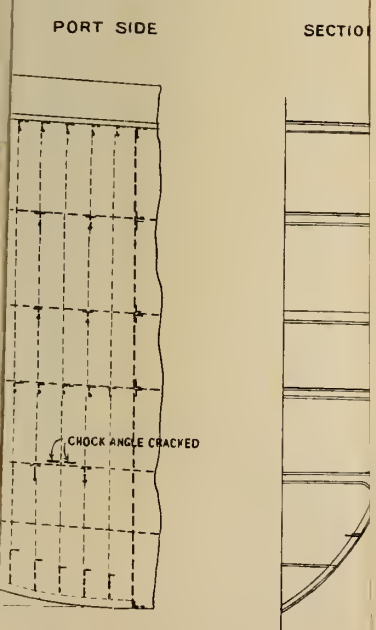
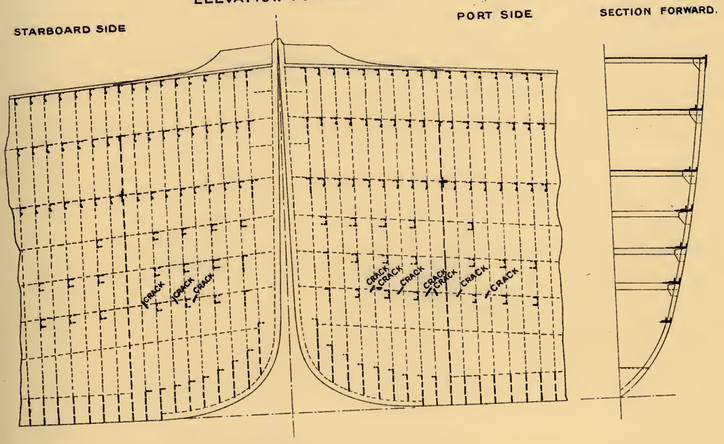
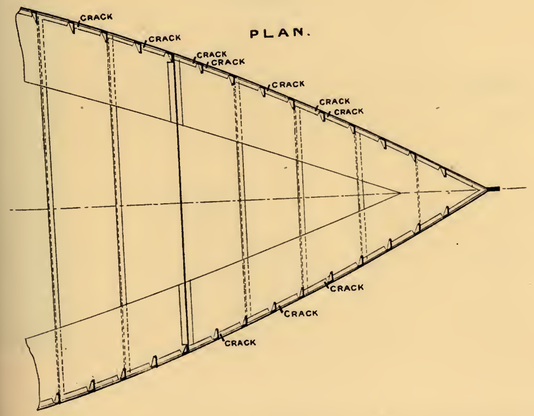


To Illustrate Dr. S. F. P. Thearle's Paper, "On Some Cases of 'Fatigue' in the Steel Material of Steamers."

Fig. 1.





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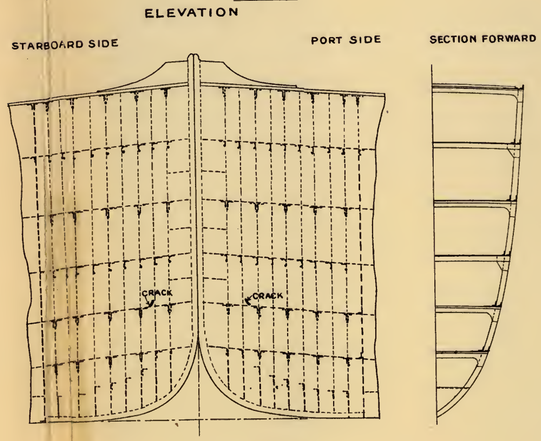

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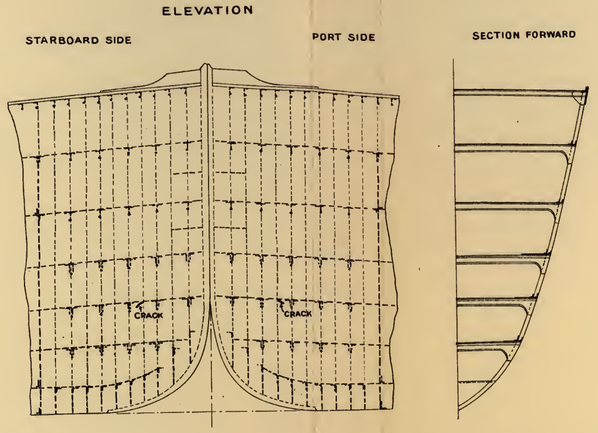

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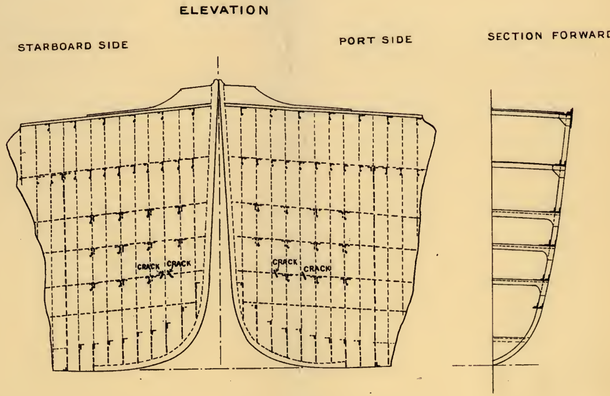

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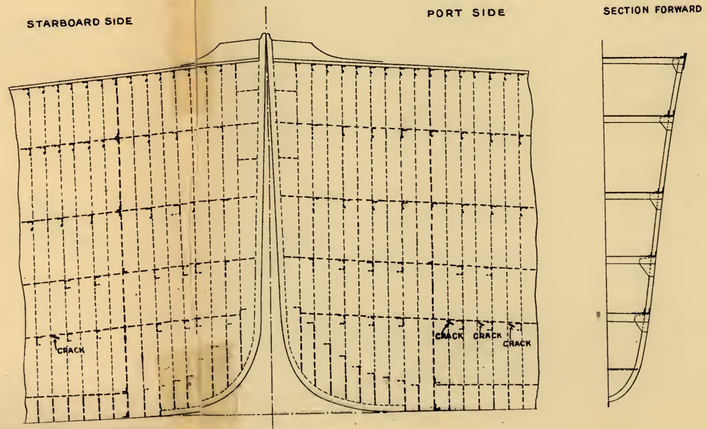

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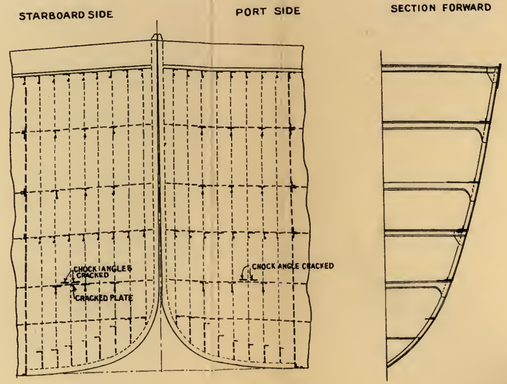

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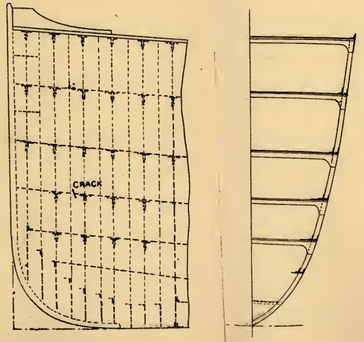

To Illustrate Dr. S. J. P. Thearle's Paper, "On Some Cases of 'Fatigue' in the Steel Material of Steamers."

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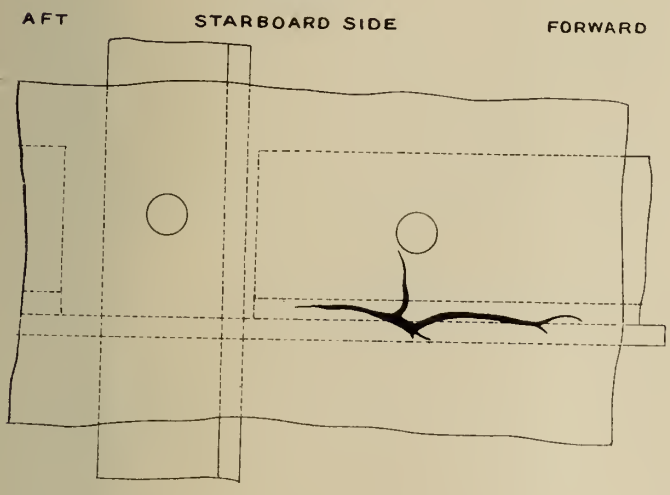

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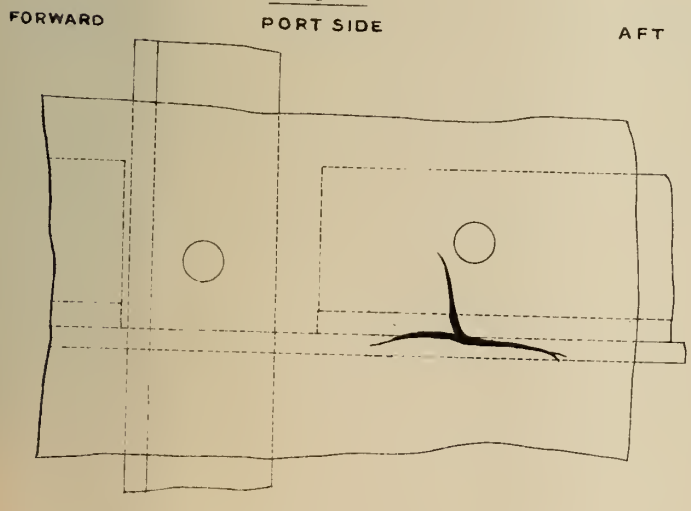

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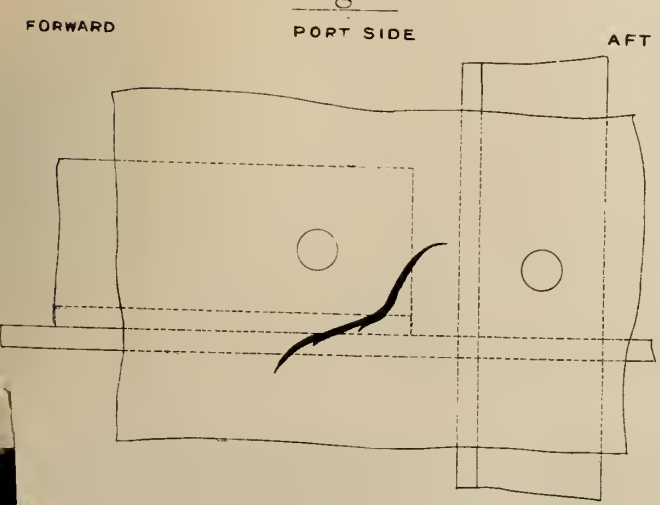
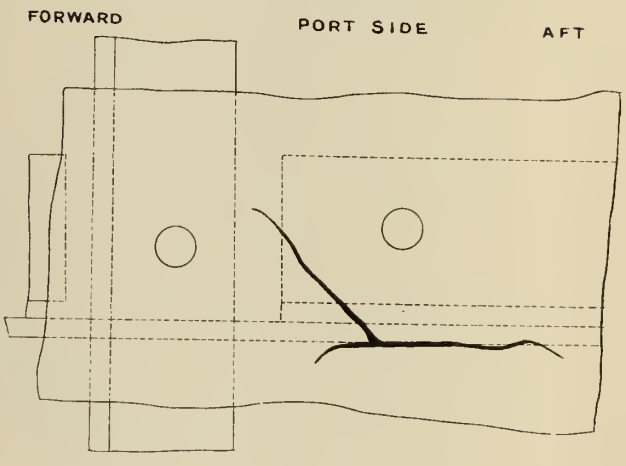


Fig. 11.



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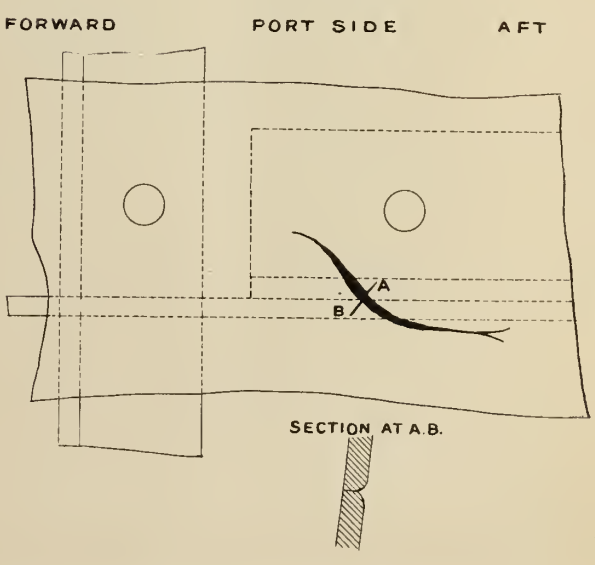

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Fig. 3. (§ 11)

MODELS

F ENTRANCE.

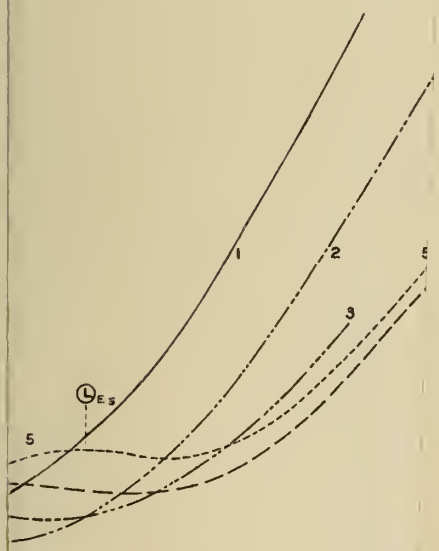


Fig. 1 (§ 4 & 5)

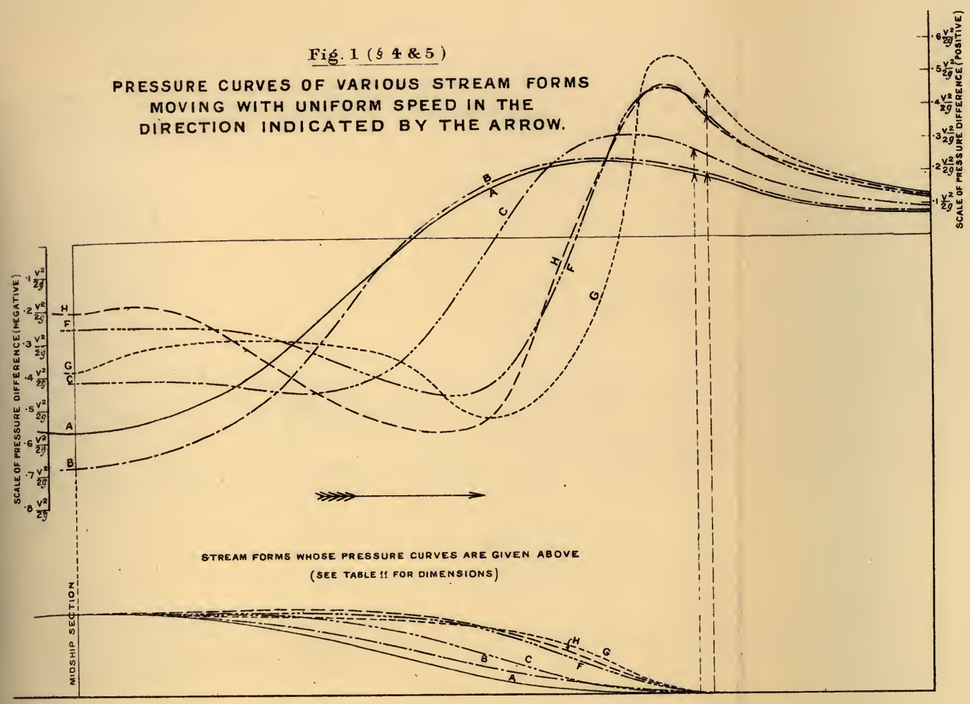
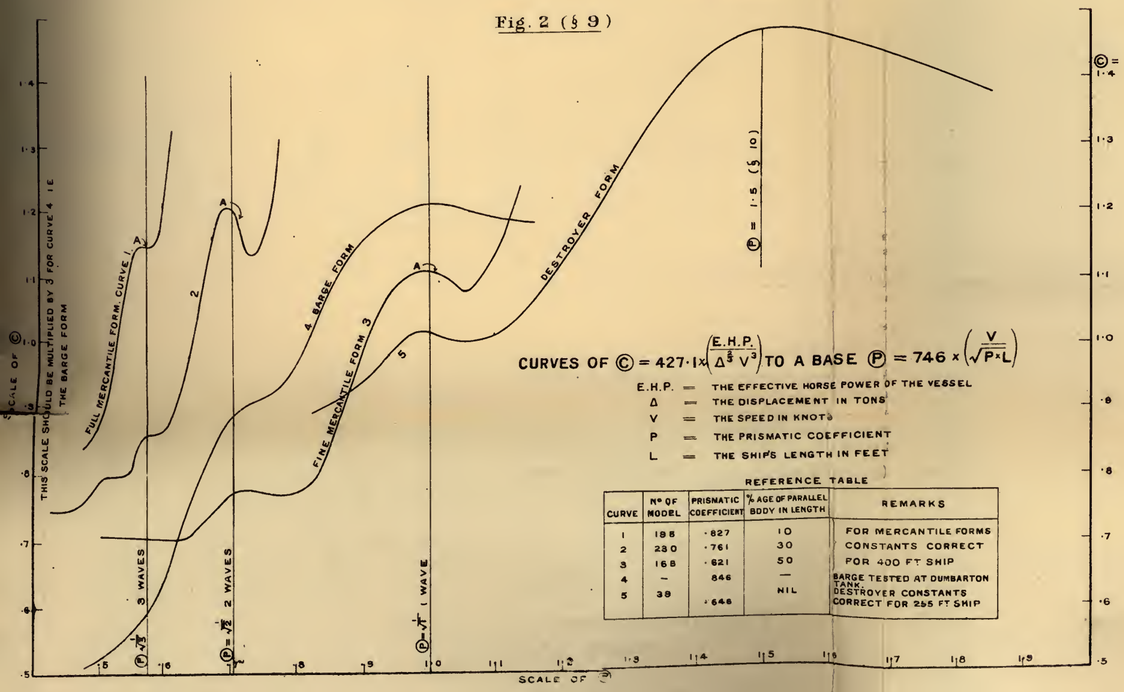

PRESSURE CURVES OF VARIOUS STREAM FORMS
MOVING WITH UNIFORM SPEED IN THE
DIRECTION INDICATED BY THE ARROW.
SCALE OF PRESSURE DIFFERENCE (NEGATIVE)
SCALE OF PRESSURE DIFFERENCE (POSITIVE)
STREAM FORMS WHOSE PRESSURE CURVES ARE GIVEN ABOVE
(SEE TABLE II FOR DIMENSIONS)
MIDSHIP SECTION

Fig. 2 (§ 9)


FULL MERCANTILE FORM. CURVE 1
FINE MERCANTILE FORM 3
4 BARGE FORM
DESTROYER FORM
THIS SCALE SHOULD BE MULTIPLIED BY 3 FOR CURVE 4 I.E. THE BARGE FORM
3 WAVES
2 WAVES
1 WAVE
SCALE OF Ⓒ
SCALE OF Ⓚ

CURVES OF Ⓒ = $427 \cdot 1 \times \left(\frac{E.H.P.}{\Delta^{\frac{2}{3}} V^3}\right)$ TO A BASE Ⓟ = $746 \times \left(\frac{V}{\sqrt{P \times L}}\right)$

E.H.P. = THE EFFECTIVE HORSE POWER OF THE VESSEL
Δ = THE DISPLACEMENT IN TONS
V = THE SPEED IN KNOTS
P = THE PRISMATIC COEFFICIENT
L = THE SHIP'S LENGTH IN FEET

REFERENCE TABLE

CURVE	Nº OF MODEL	PRISMATIC COEFFICIENT	% AGE OF PARALLEL BODY IN LENGTH	REMARKS
1	18 S	·827	10	FOR MERCANTILE FORMS CONSTANTS CORRECT FOR 400 FT SHIP
2	23 O	·761	30	
3	16 B	·621	50	
4	—	·846	—	BARGE TESTED AT DUMBARTON TANK.
5	38	·648	NIL	DESTROYER CONSTANTS CORRECT FOR 255 FT SHIP

Fig. 3. (§ 11)

CURVES OF Ⓒ FOR A SERIES OF MODELS

HAVING THE SAME TOTAL LENGTH BUT VARYING LENGTH OF ENTRANCE.
CORRECTED FOR A SHIP OF 400 FEET LENGTH

REFERENCE TABLE

Nº OF CURVE	Nº OF MODEL	LENGTH OF ENTRANCE / LENGTH OF RUN	PRISMATIC COEFFICIENT	Ⓚ FOR SPEED V WHERE $\frac{V}{\sqrt{L_E}} = \text{Ⓛ}_E = 1 \cdot 095$
1	17 B	·566	·623	1·61
2	17 A	·684	·623	1·71
3	14 B	·804	·622	1·85
4	16 B	1·214	·621	1·98
5	16 O	1·624	·620	2·03

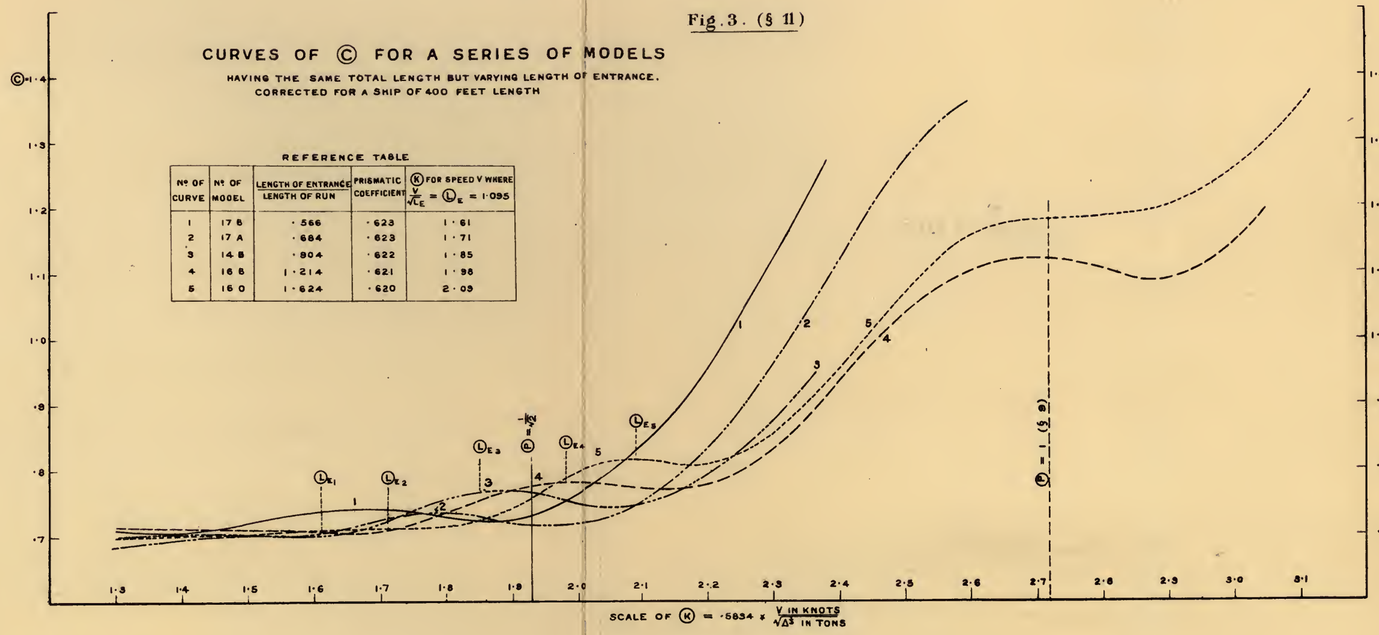


Fig. 4. (§ 11)

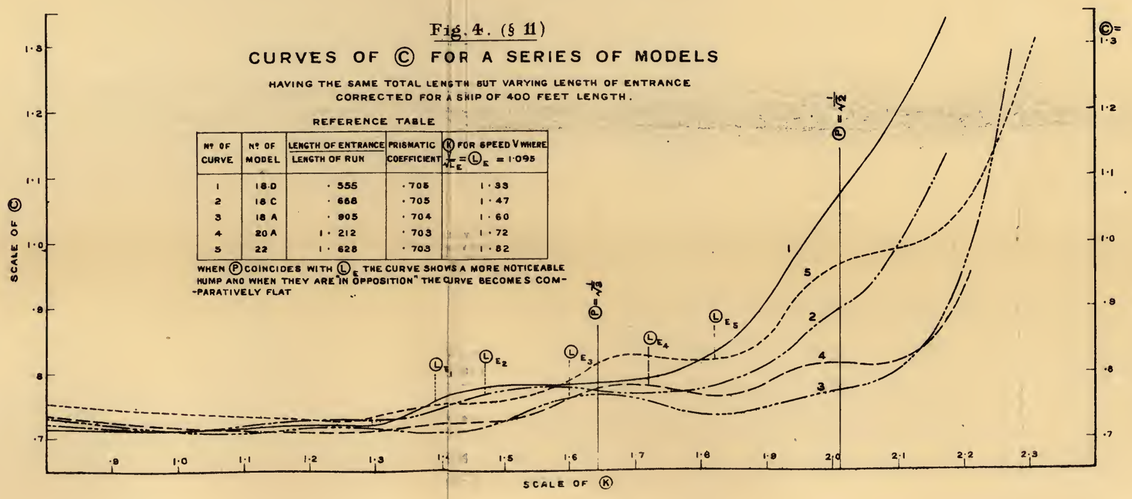
CURVES OF Ⓒ FOR A SERIES OF MODELS

HAVING THE SAME TOTAL LENGTH BUT VARYING LENGTH OF ENTRANCE
CORRECTED FOR A SHIP OF 400 FEET LENGTH.

REFERENCE TABLE

Nº OF CURVE	Nº OF MODEL	LENGTH OF ENTRANCE / LENGTH OF RUN	PRISMATIC COEFFICIENT	Ⓚ FOR SPEED V WHERE $\frac{V}{\sqrt{L_E}} = \text{Ⓛ}_E = 1 \cdot 095$
1	18 D	·555	·705	1·33
2	18 C	·668	·705	1·47
3	18 A	·805	·704	1·60
4	20 A	1·212	·703	1·72
5	22	1·628	·703	1·82

WHEN Ⓟ COINCIDES WITH Ⓛ_E THE CURVE SHOWS A MORE NOTICEABLE HUMP AND WHEN THEY ARE "IN OPPOSITION" THE CURVE BECOMES COMPARATIVELY FLAT



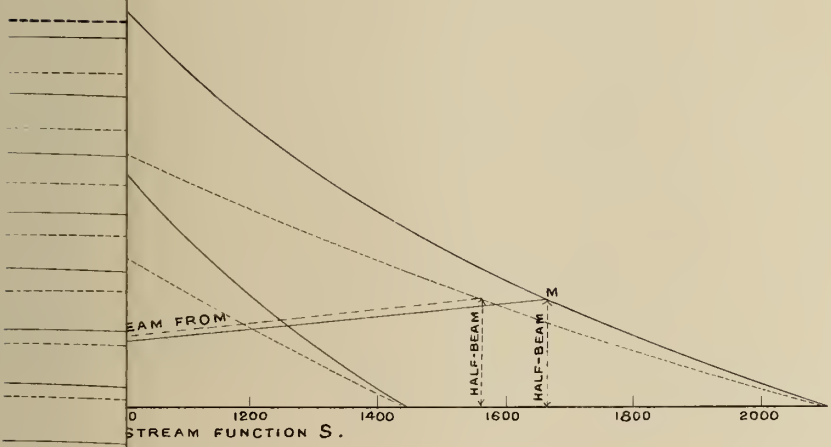
nce of Ships."

. (Appendix 1.)

STREAM FUNCTIONS

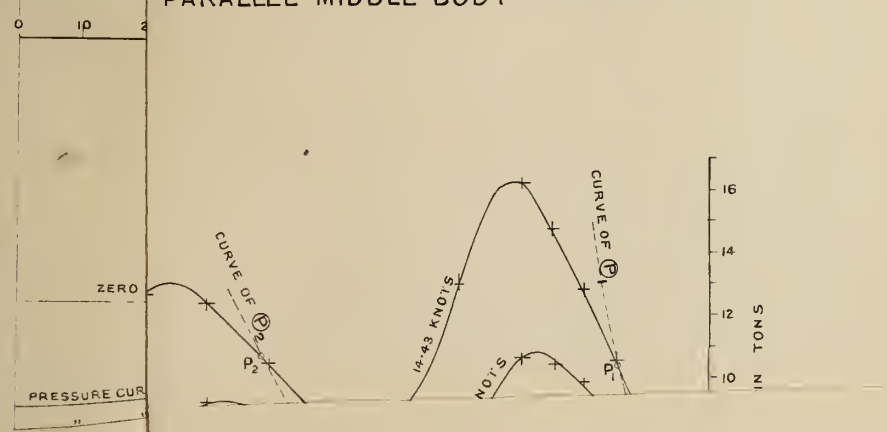
——— IN UNLIMITED FLUID
- - - - - IN RESTRICTED FLUID

H LINE REPRESENTS THE VARIATION OF STREAM FUNCTION ALONG THE ORDINATE
HE LONGITUDINAL DISTANCE FROM AMIDSHIPS GIVEN BY THE NUMBER ON IT
ACTUAL VALUE OF S IS OBTAINED BY MEASURING TO THE LINE FROM THE AXIS A B
HE DISTANCE OUT GIVEN BY THE ORDINATE.



g. 8 (§ 14)

STANCE OBTAINED BY W. FROUDE FOR
PARALLEL MIDDLE BODY



To Illustrate Messrs. G. S. Baker and J. L. Kent's Paper on "The Effect of Form and Size on the Resistance of Ships."

Fig. 5 (§ 6 and App. II.)

STREAM LINES AND PRESSURE CURVES FOR ANGULAR FORM

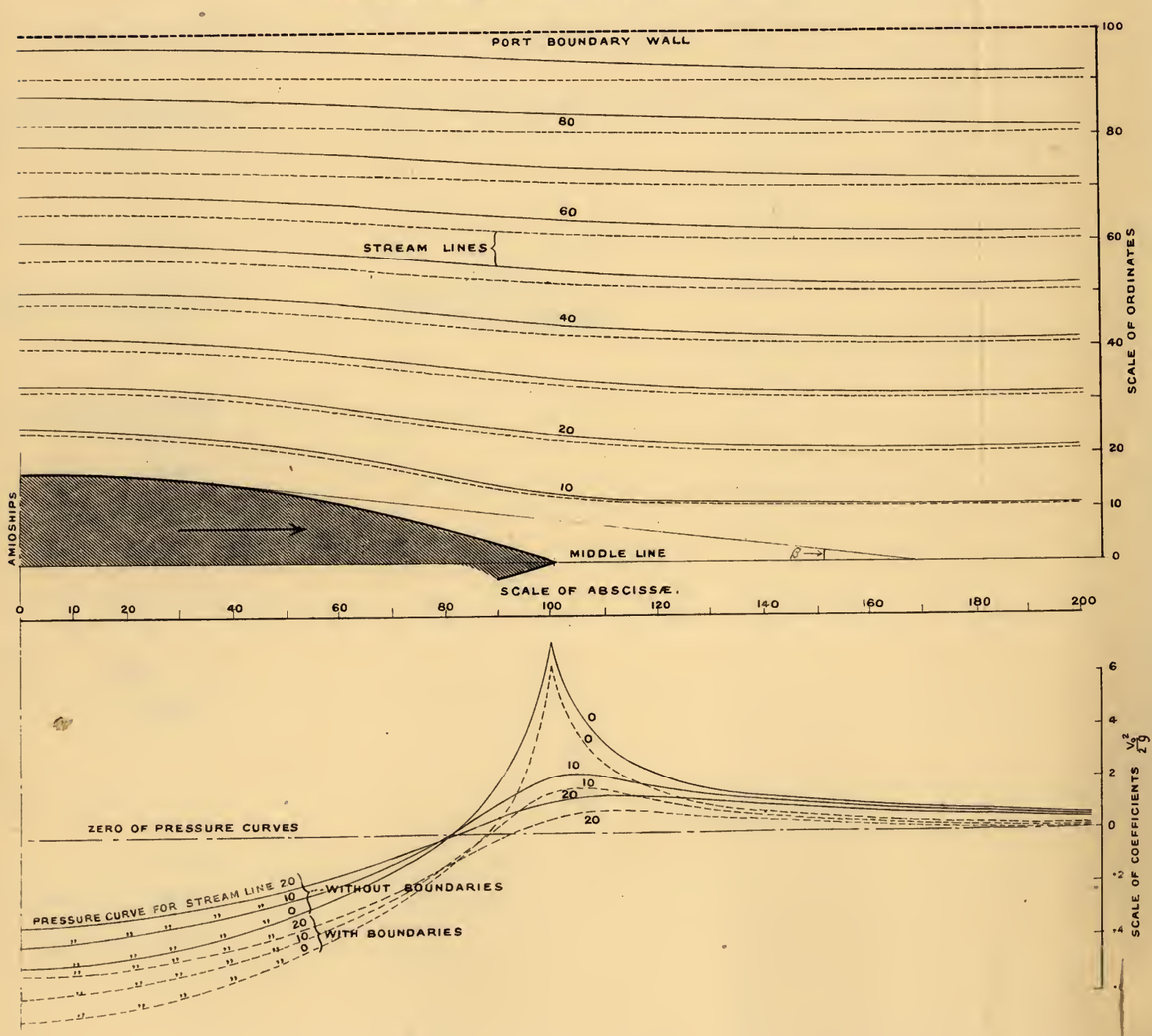


Fig. 7. (Appendix 1.)

CURVES OF STREAM FUNCTIONS

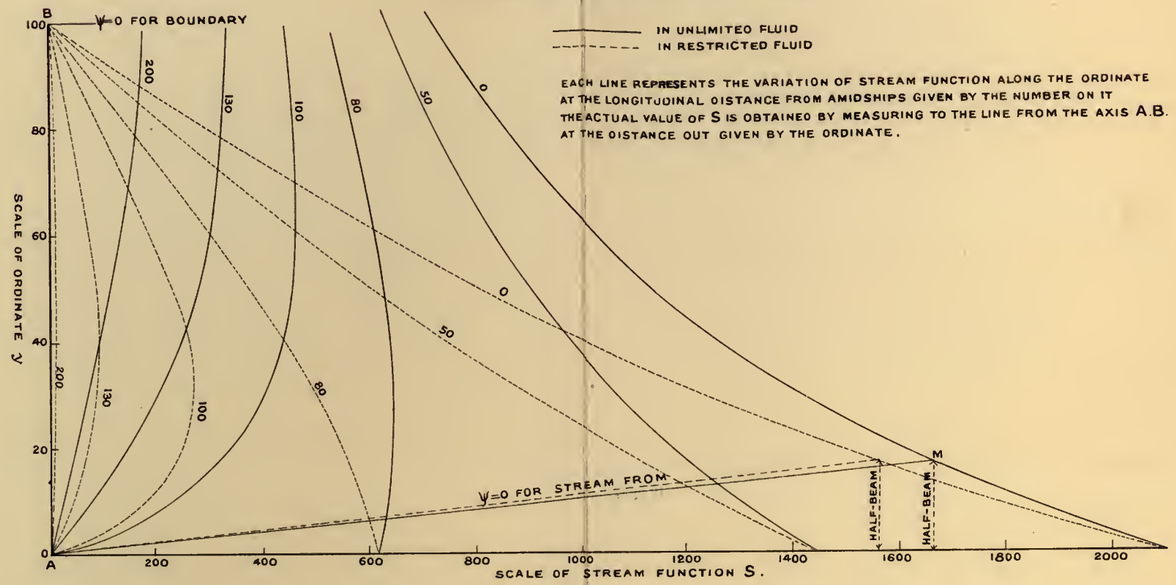
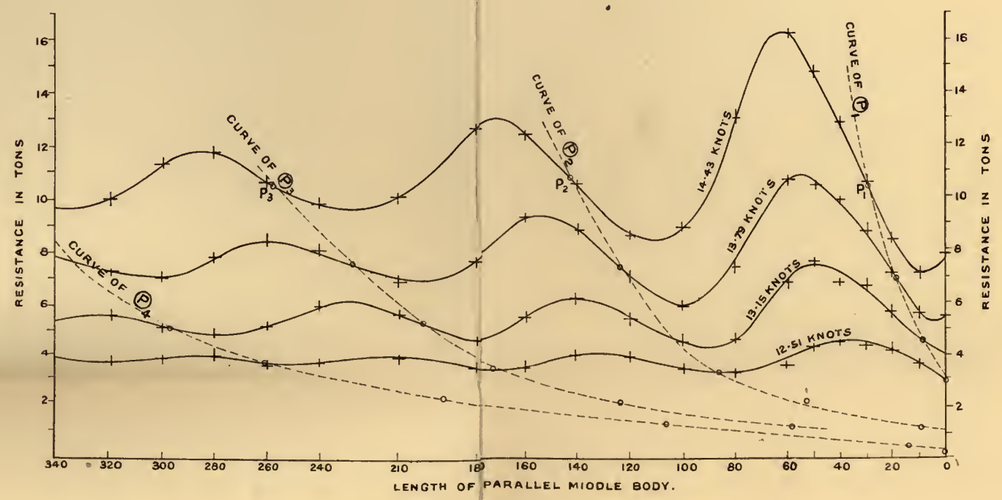
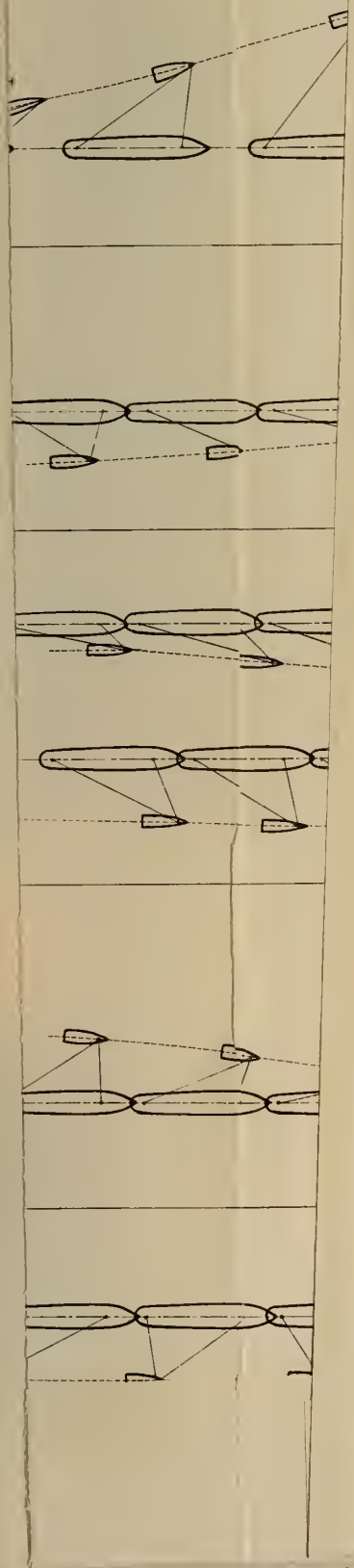

IN UNLIMITED FLUID
IN RESTRICTED FLUID
EACH LINE REPRESENTS THE VARIATION OF STREAM FUNCTION ALONG THE ORDINATE AT THE LONGITUDINAL DISTANCE FROM AMIDSHIPS GIVEN BY THE NUMBER ON IT THE ACTUAL VALUE OF S IS OBTAINED BY MEASURING TO THE LINE FROM THE AXIS A.B. AT THE DISTANCE OUT GIVEN BY THE ORDINATE.
B
ψ=0 FOR BOUNDARY
200
150
100
80
50
0
A
ψ=0 FOR STREAM FROM
M
HALF-BEAM
HALF-BEAM
SCALE OF ORDINATE y
0 20 40 60 80 100
SCALE OF STREAM FUNCTION S.
200 400 600 800 1000 1200 1400 1600 1800 2000

Fig. 8 (§ 14)

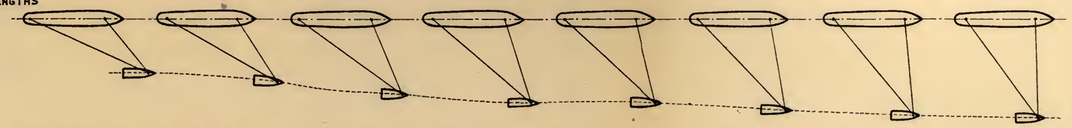
CURVES OF RESIDUARY RESISTANCE OBTAINED BY W. FROUDE FOR VARYING LENGTHS OF PARALLEL MIDDLE BODY


CURVE OF P1
CURVE OF P2
CURVE OF P3
CURVE OF P4
14·43 KNOTS
13·79 KNOTS
13·15 KNOTS
12·51 KNOTS
RESISTANCE IN TONS
0 2 4 6 8 10 12 14 16
340 320 300 280 260 240 210 180 160 140 120 100 80 60 40 20 0
LENGTH OF PARALLEL MIDDLE BODY.



To Illustrate Professor A. H. Gibson and Mr. J. Hannay Thompson's Paper, "Experiments on 'Suction,' or Interaction between Passing Vessels."

(1) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 1⅝ LENGTHS



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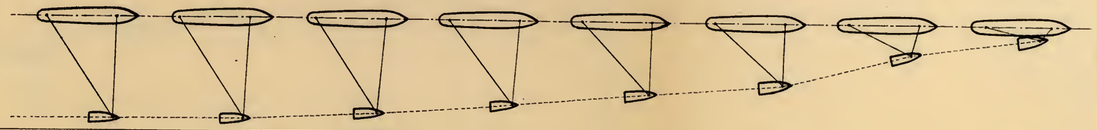
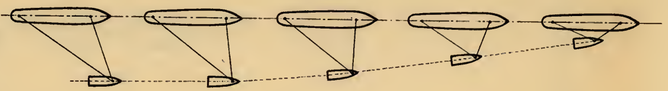
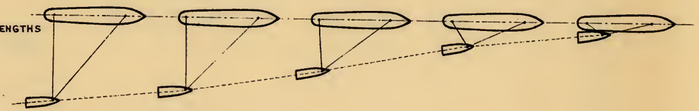


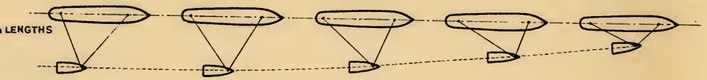
(2) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 2 LENGTHS



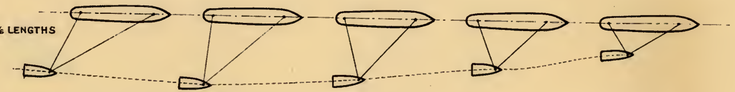
(3) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 2⅔ LENGTHS



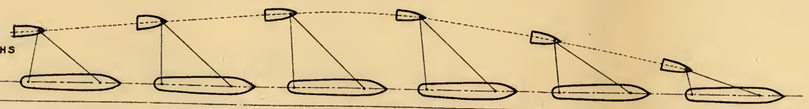
(4) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 1⅓ LENGTHS



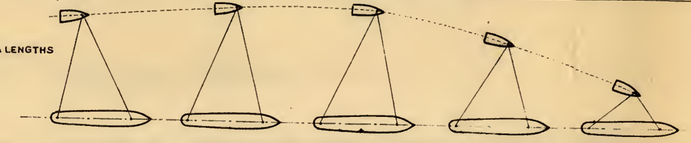
(5) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 1⅔ LENGTHS



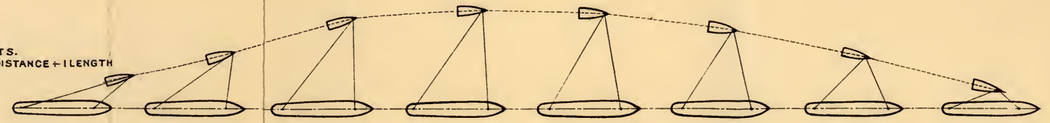
(6) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 1⅔ LENGTHS



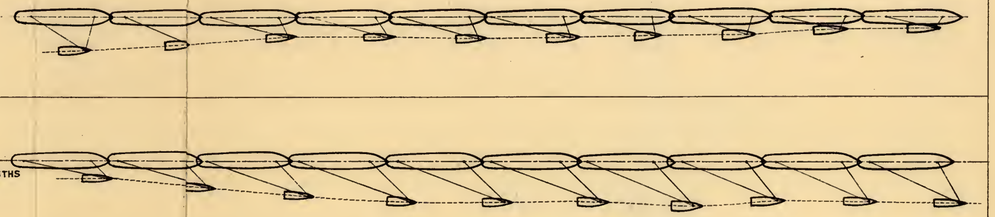
(7) SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 3⅓ LENGTHS



(8) SPEED 5 KNOTS.
INITIAL LATERAL DISTANCE 1 LENGTH



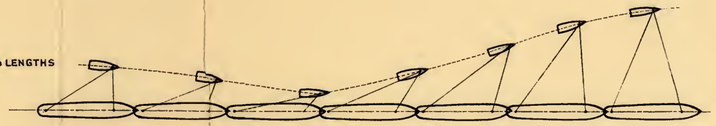
(9) SPEED 3·6 KNOTS.
INITIAL LATERAL DISTANCE
1⅒ LENGTHS.



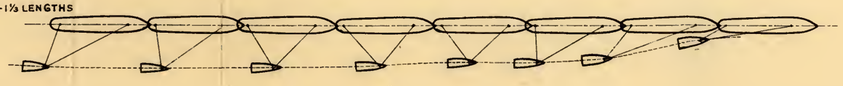
(10) SPEED 3·6 KNOTS.
INITIAL LATERAL DISTANCE ⅔ LENGTHS

10 CONTINUED.

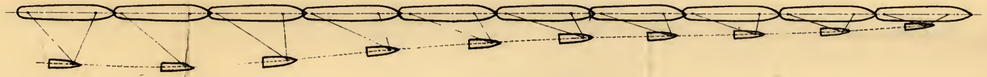
(11) SPEED 3·6 KNOTS.
INITIAL LATERAL DISTANCE 1⅓ LENGTHS

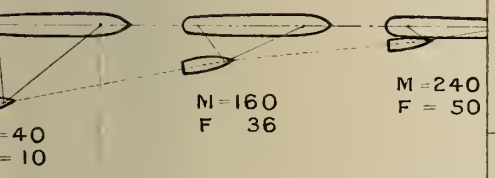


(12) SPEED 3·6 KNOTS.
INITIAL LATERAL DISTANCE 1⅓ LENGTHS



(13) SPEED 3·6 KNOTS.
INITIAL LATERAL DISTANCE 1⅓ LENGTHS

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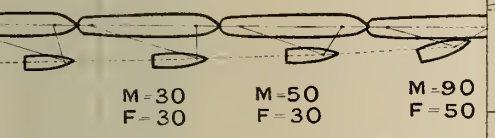
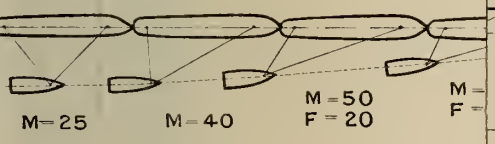
F 36
M=240
F = 50
=40
= 10


M=30
F=30
M=50
F=30
M=90
F=50

2°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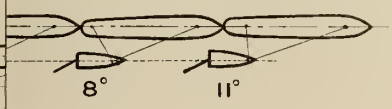
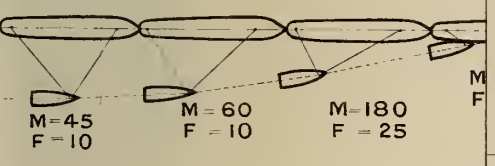
M=40
M=50
F=20


8°
11°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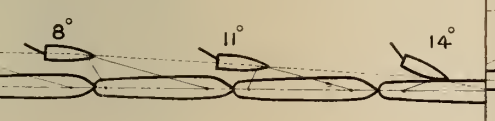
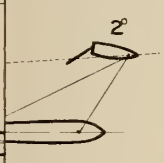
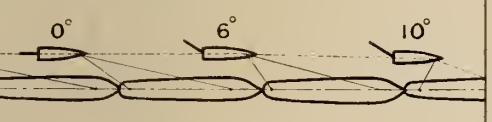
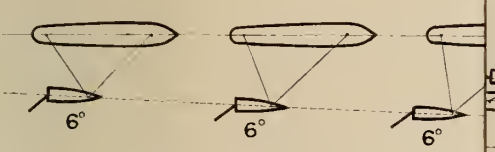
F=10
M=60
F=10
M=180
F=25

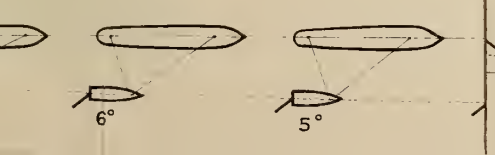

8°
11°
14°

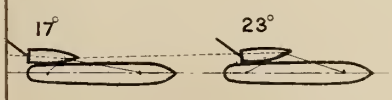

2°


0°
6°
10°


6°
6°
6°

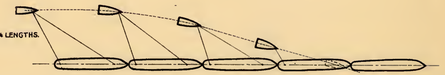
8°
10°
13°


6°
5°

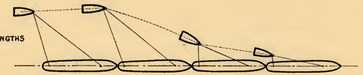

17°
23°

To Illustrate Professor A. H. Gibson and Mr. J. Hannay Thompson's Paper, "Experiments on 'Suction,' or Interaction between Passing Vessels."

(14)
SPEED 3·75 KNOTS
INITIAL LATERAL DISTANCE >2¼ LENGTHS.



(15)
SPEED 3·75 KNOTS.
INITIAL LATERAL DISTANCE >2¼ LENGTHS



(16)
SPEED 3·75 KNOTS
INITIAL LATERAL DISTANCE >1½ LENGTHS

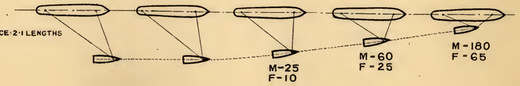
(17)
SPEED 3·75 KNOTS
INITIAL LATERAL DISTANCE > 2 LENGTHS.

RAN PAST WITHOUT COLLISION

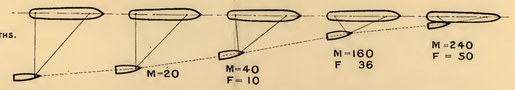
(18)
SPEED 5·8 KNOTS
INITIAL LATERAL DISTANCE: 1½ LENGTHS

(19)
SPEED 5·8 KNOTS
INITIAL LATERAL DISTANCE: 2 LENGTHS

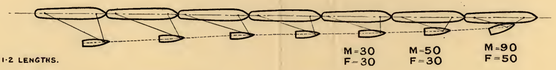
(20)
SPEED 3 KNOTS.
INITIAL LATERAL DISTANCE 2·1 LENGTHS



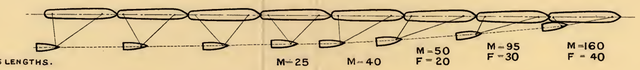
(21)
SPEED 5 KNOTS
INITIAL LATERAL DISTANCE 2·7 LENGTHS.



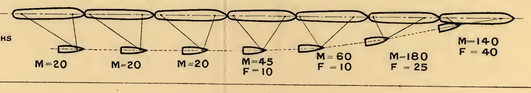
(22)
SPEED 3·61 KNOTS
INITIAL LATERAL DISTANCE 1·2 LENGTHS.



(23)
SPEED 3·61 KNOTS
INITIAL LATERAL DISTANCE 1·55 LENGTHS.



(24)
SPEED 3·61 KNOTS
INITIAL LATERAL DISTANCE 1·52 LENGTHS



(25)
SPEEDS {P.L. 3·6 KNOTS
M.B. 4·0 KNOTS

INITIAL LATERAL DISTANCE ·7 LENGTH.

(26)
SPEEDS {P.L. 3·6 KNOTS
M.B. 4·0 KNOTS

INITIAL LATERAL DISTANCE ·8 LENGTH

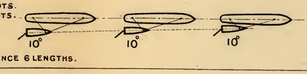
(27)
SPEEDS {P.L. 5·0 KNOTS
M.B. 4·5 KNOTS

INITIAL LATERAL DISTANCE 1·05 LENGTHS

(28)
SPEEDS {P.L. 5·0 KNOTS
M.B. 5·2 KNOTS

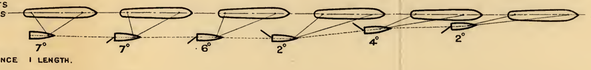
INITIAL LATERAL DISTANCE ·27 LENGTH

(29)
SPEEDS {P.L. 5·0 KNOTS.
M.B. 5·0 KNOTS.



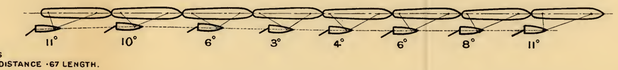
INITIAL LATERAL DISTANCE 6 LENGTHS.

(30)
SPEEDS {P.L. 4·5 KNOTS
M.B. 3·6 KNOTS



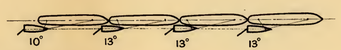
INITIAL LATERAL DISTANCE 1 LENGTH.

(31)



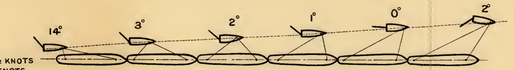
SPEED 3½ KNOTS
INITIAL LATERAL DISTANCE ·67 LENGTH.

(32)



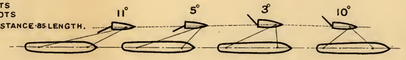
SPEED 3½ KNOTS
INITIAL LATERAL DISTANCE ·33 LENGTH.

(33)



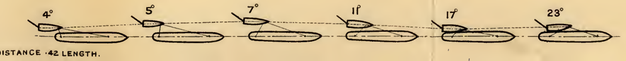
SPEEDS {P.L. 3½ KNOTS
M.B. 5 KNOTS
INITIAL LATERAL DISTANCE ·44 LENGTH.

(34)
SPEEDS {P.L. 5 KNOTS
M.B. 3½ KNOTS
INITIAL LATERAL DISTANCE ·85 LENGTH.



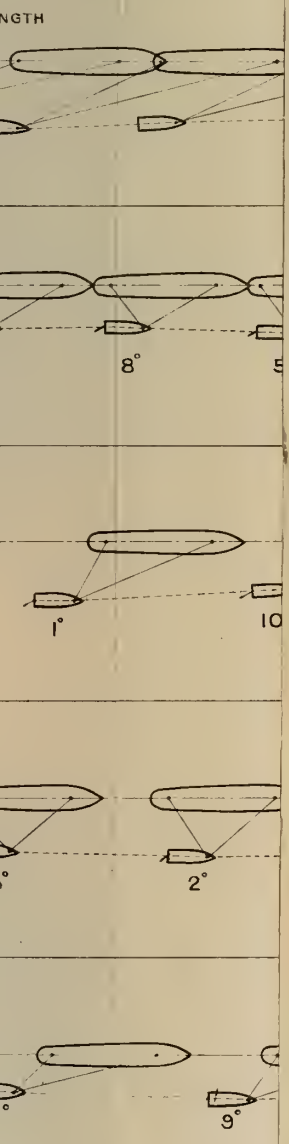
(35)
SPEEDS {P.L. 5 KNOTS
M.B. 3·6 KNOTS
INITIAL LATERAL DISTANCE ·68 LENGTH.

(36)

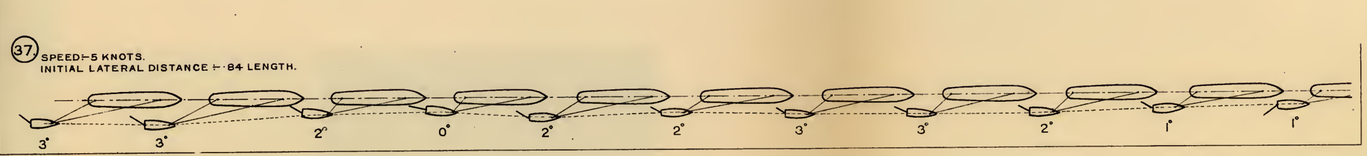


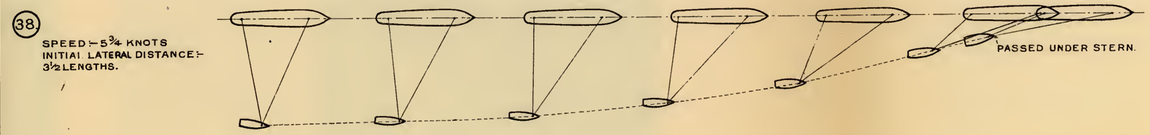
SPEED 5 KNOTS
INITIAL LATERAL DISTANCE ·42 LENGTH.

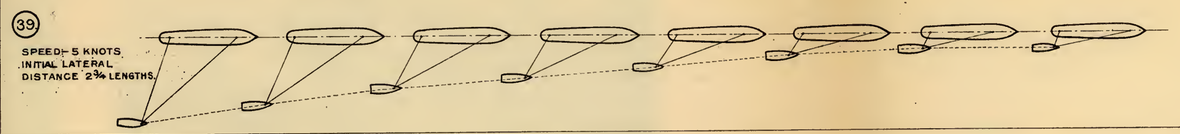
tion between Passin

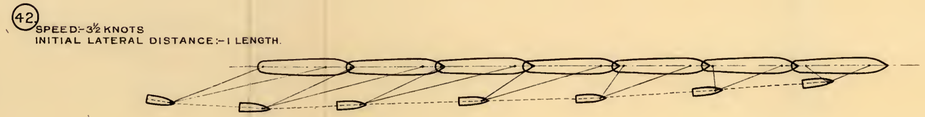


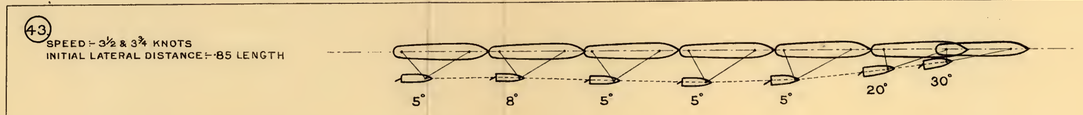
To Illustrate Professor A. H. Gibson and Mr. J. Hannay Thompson's Paper, "Experiments on 'Suction,' or Interaction between Passing Vessels."

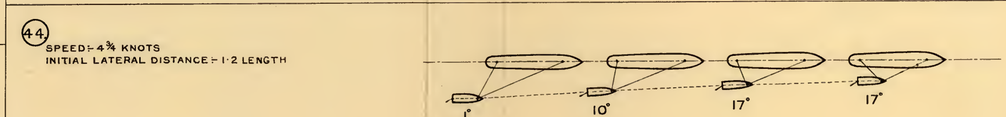


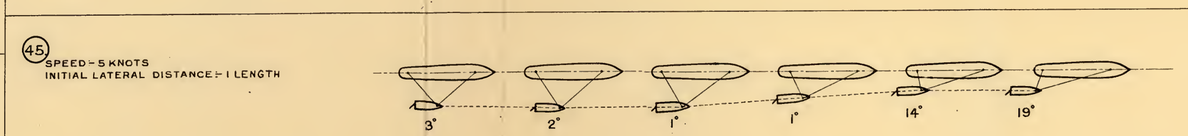


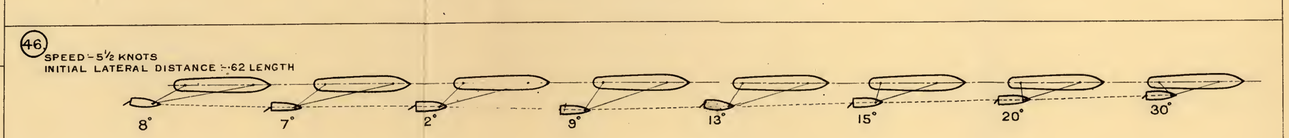












To Illustrate Mr. A. Cannon's Paper on "Experimental Determination of the Effect of Internal Loose Water upon the Rolling of a Ship amongst a Regular Series of Waves."

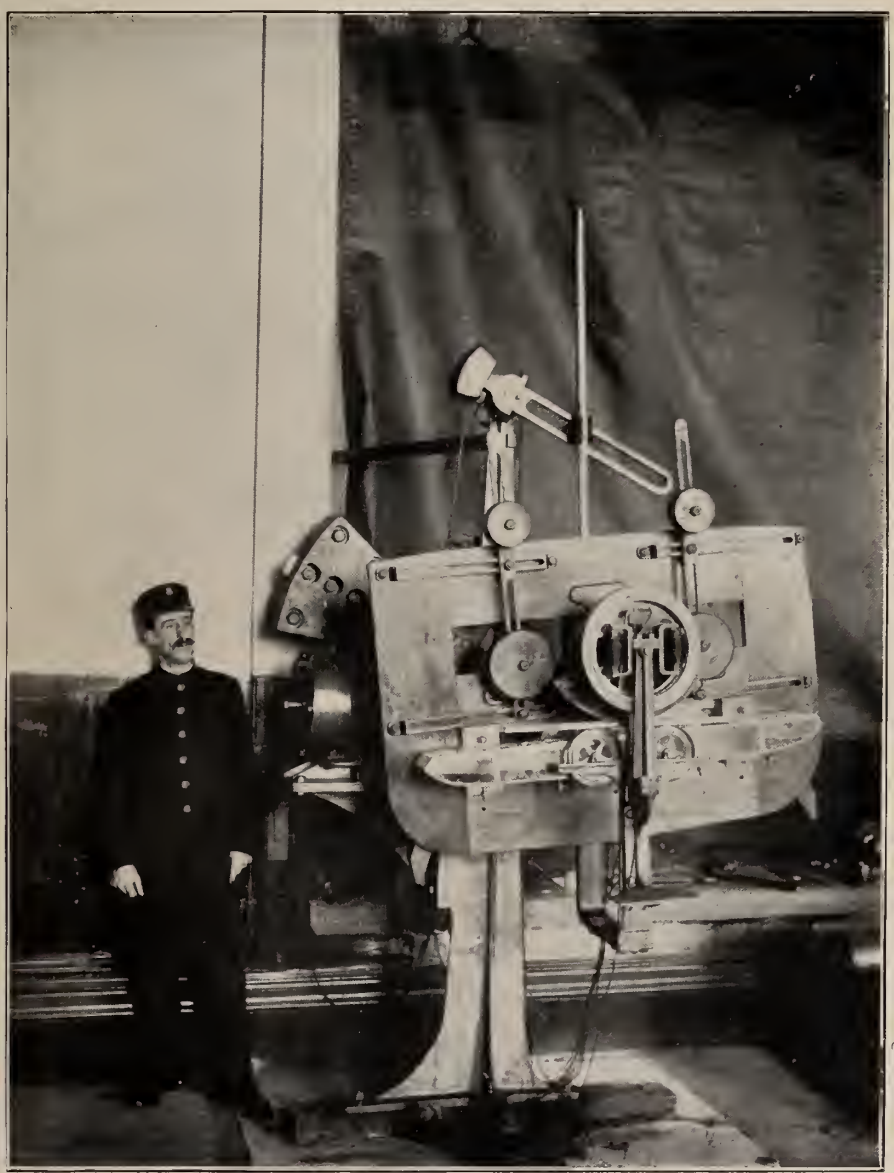
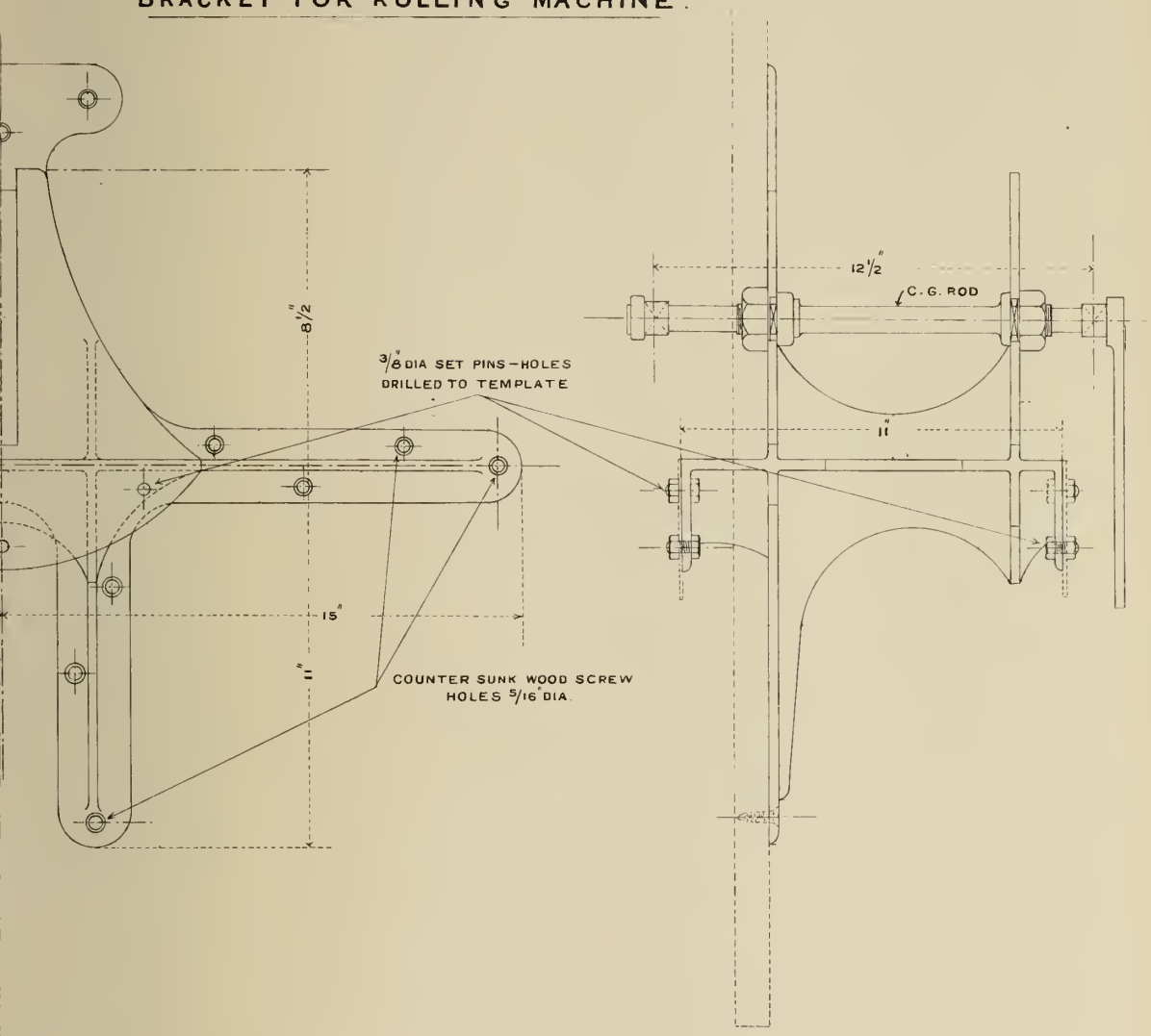


FIG. 4.

g of a Ship amongst a Regular Series of Waves."

Figs. 5 & 6.

BRACKET FOR ROLLING MACHINE.



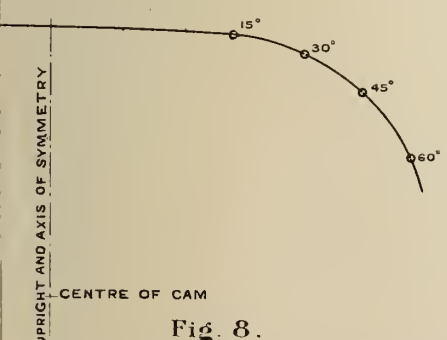


Fig. 8.

CAM FOR EQUILATERAL TRIANGULAR PRISM, FLOATING WITH VERTEX DOWNWARDS AT 50% TOTAL DISPLACEMENT.

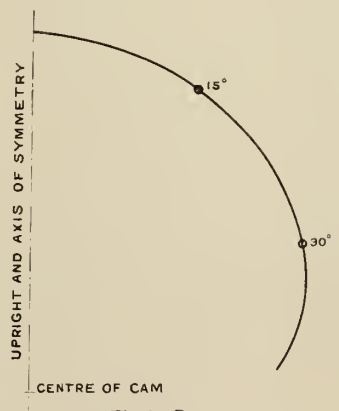


Fig. 9.

SHAPE OF CAM FOR A RECTANGULAR PRISM OF BREADTH = TWICE DEPTH AT 50% DISPLACEMENT

To Illustrate Mr. A. Cannon's Paper on "Experimental Determination of the Effect of Internal Loose Water upon the Rolling of a Ship amongst a Regular Series of Waves."

Fig. 2.

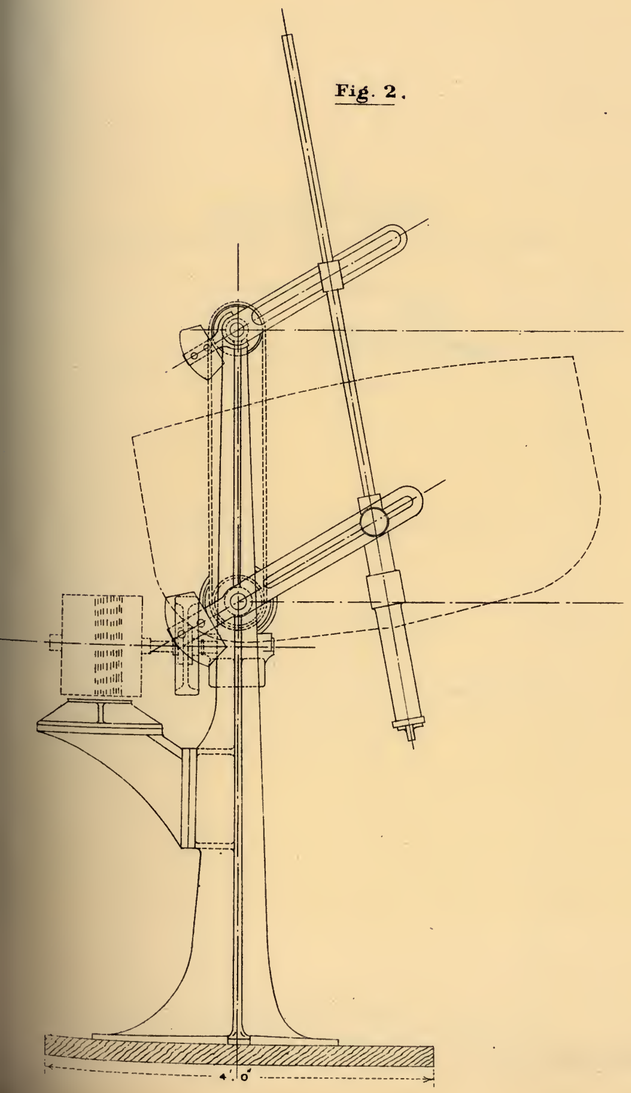
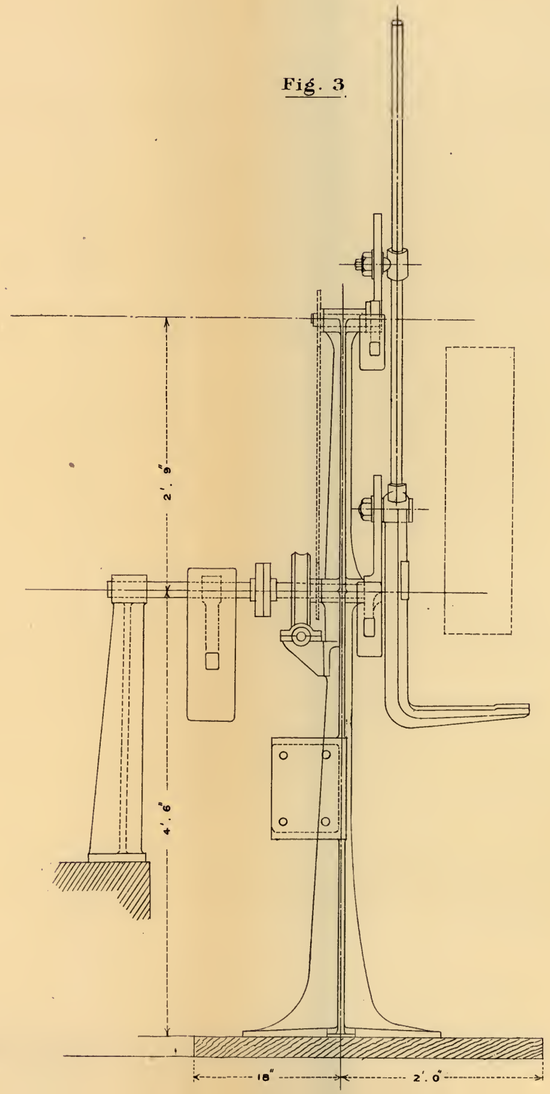
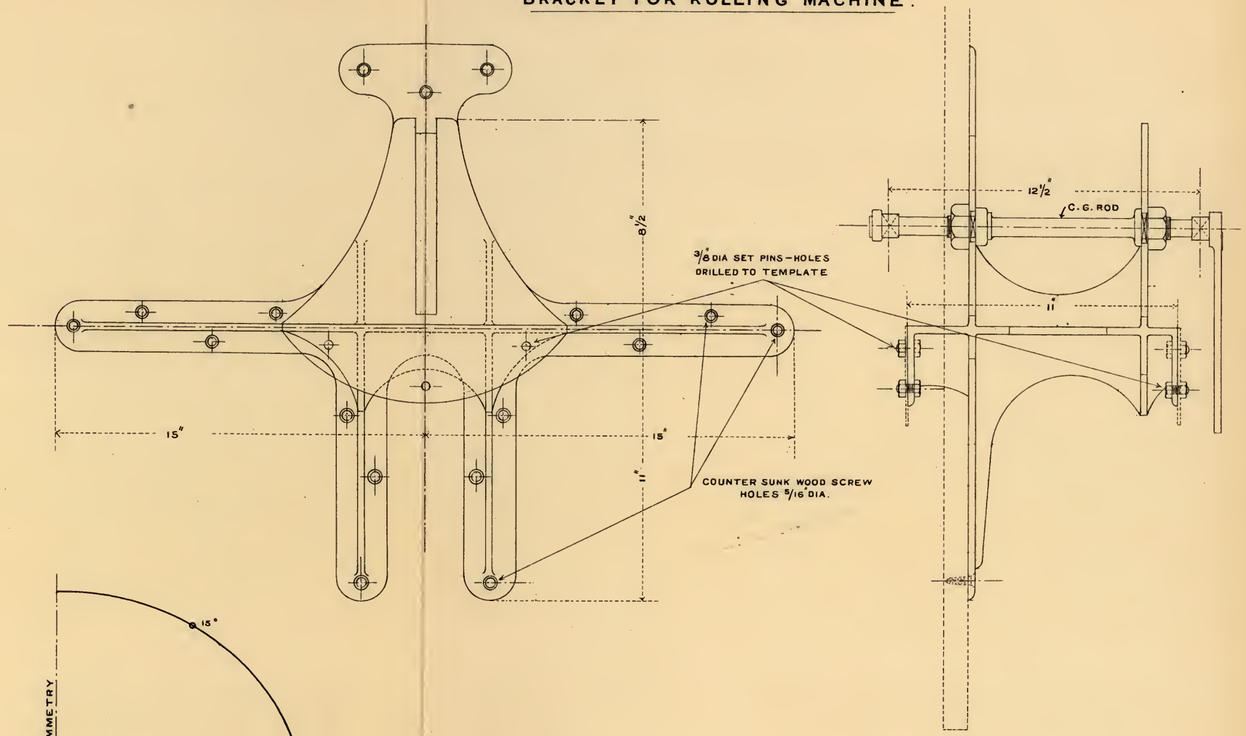


Fig. 3



Figs. 5 & 6.

BRACKET FOR ROLLING MACHINE.



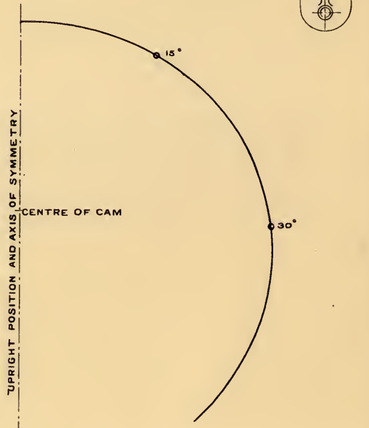


Fig. 7.

SHAPE OF CAM FOR CARGO BOAT 350′ × 45′ × 27′ AT 22′ DRAUGHT.

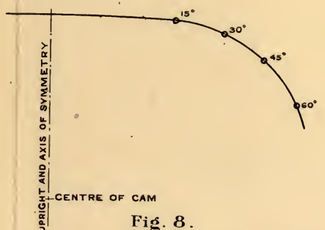


Fig. 8.

CAM FOR EQUILATERAL TRIANGULAR PRISM, FLOATING WITH VERTEX DOWNWARDS AT 50% TOTAL DISPLACEMENT.

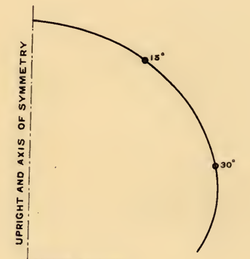


Fig. 9.

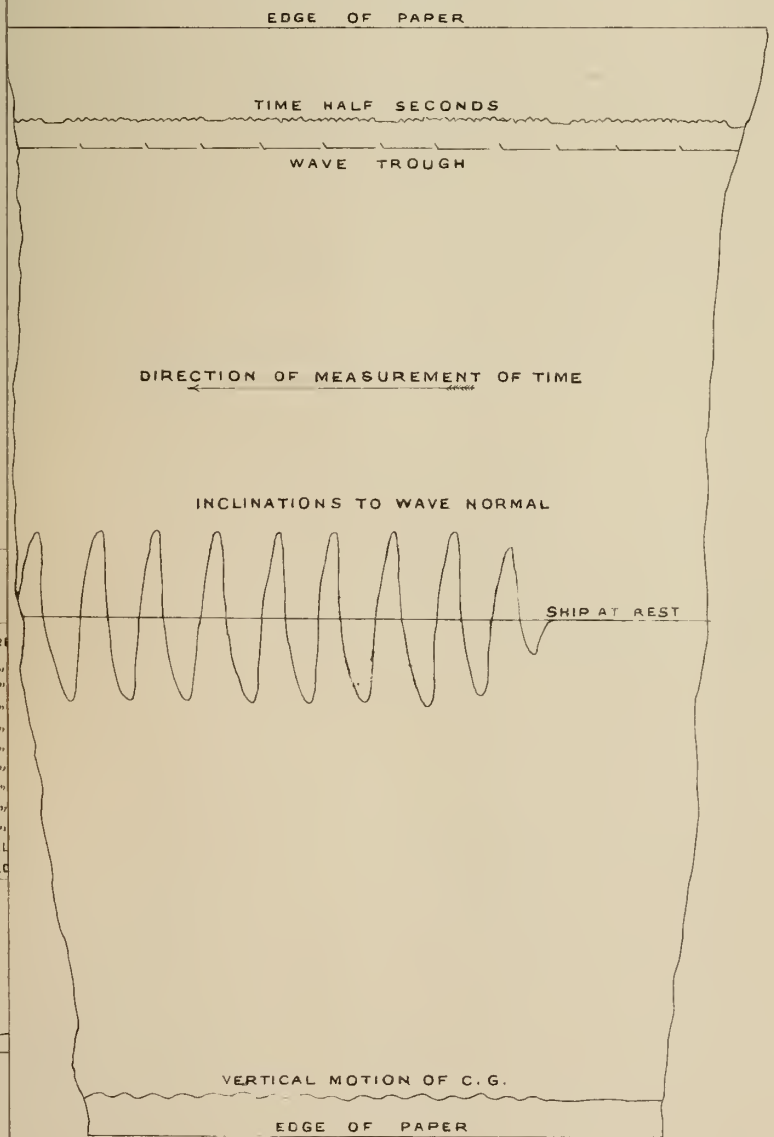
SHAPE OF CAM FOR A RECTANGULAR PRISM OF BREADTH = TWICE DEPTH AT 50% DISPLACEMENT.

amongst a Regular Series of Waves."

Fig. 13.

SPECIMEN OF RECORD

VESSEL ON WAVE 30' HIGH, 900' LONG. WITH 1·23% FREE WATER.



No. OF CURVE	
1	ND FRE
2	,, ,,
3	1·23 ,,
4	1·23 ,,
5	2·43 ,,
6	2·43 ,,
7	3·60 ,,
8	3·60 ,,
9	4·75 ,,
10	5·87 ,,
11	CHANNEL
	TAPERED

90 85

To Illustrate Mr. A. Cannon's Paper on "Experimental Determination of the Effect of Internal Loose Water upon the Rolling of a Ship amongst a Regular Series of Waves."

Fig. 10.

LOCUS OF Z FOR CARGO BOAT 182′ × 27′ × 13′.5″
AT 5′.9″ DRAUGHT AND 600 TONS DISPLACEMENT
K G = 9·8 FEET

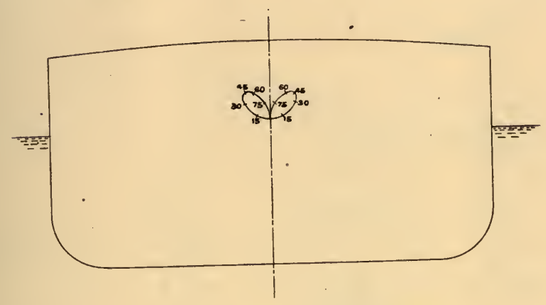


Fig. 11.

CURVES OF DECLINING ANGLES.

No. of Curve	PARTICULARS OF MODEL
1	NO FREE WATER. BRAKE BLOCKS NOT TOUCHING
2	" " " " " TOUCHING
3	1·23 " " " " NOT TOUCHING
4	1·23 " " " " TOUCHING
5	2·43 " " " " NOT TOUCHING
6	2·43 " " " " TOUCHING
7	3·60 " " " " NOT TOUCHING
8	3·60 " " " " TOUCHING
9	4·75 " " " " TOUCHING & NOT TOUCHING
10	5·87 " " " " " " " "
11	CHANNEL STEAMER WITH BILGE KEELS 90′ LONG AND 1′.6″ BROAD TAPERED AT ENDS

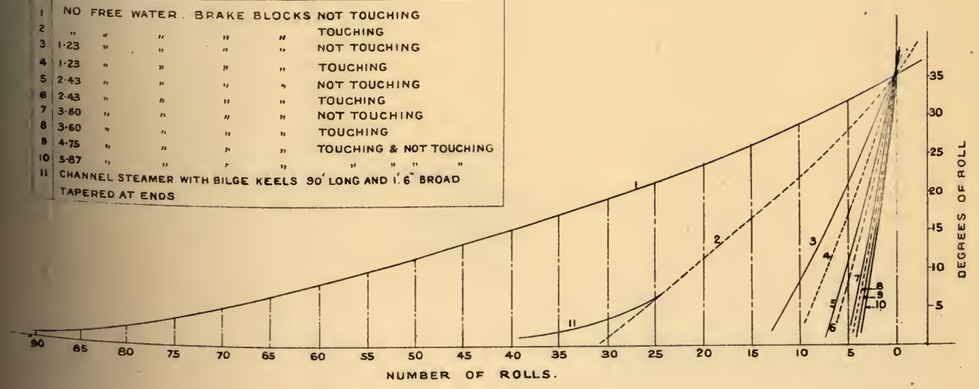
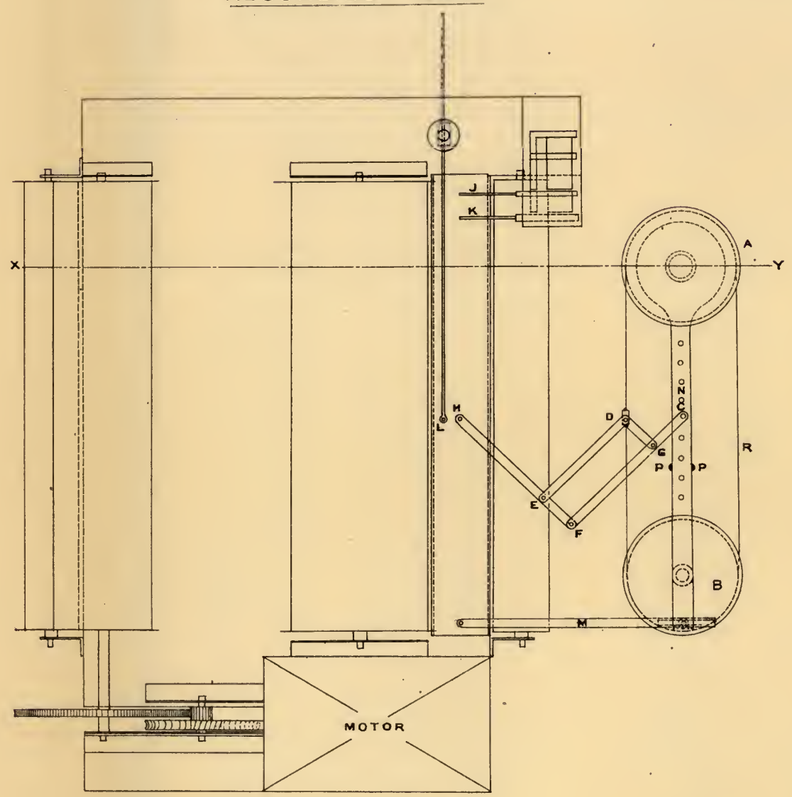
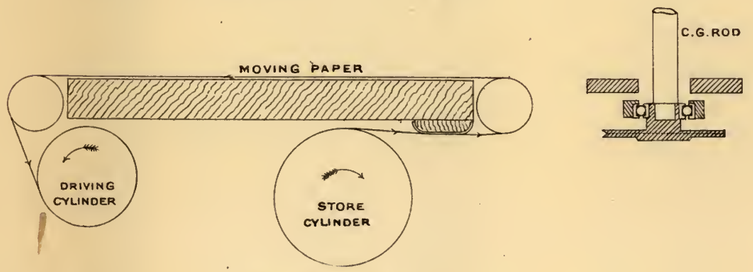


Fig. 12.

RECORDING GEAR.



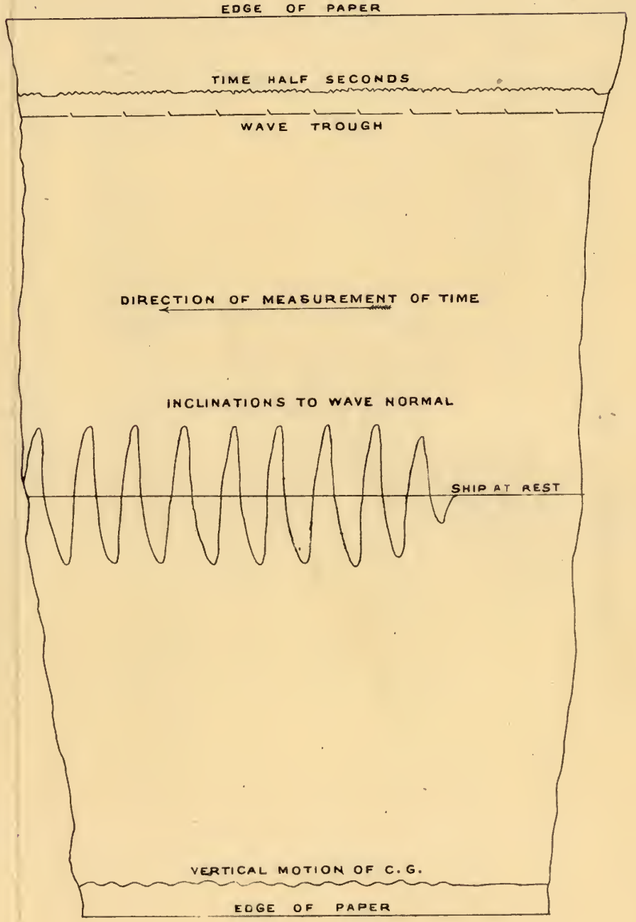


SECTION AT X.Y.

Fig. 13.

SPECIMEN OF RECORD

VESSEL ON WAVE 30′ HIGH, 900′ LONG. WITH 1·23% FREE WATER.



...E SURFACE

...TING
...F THE

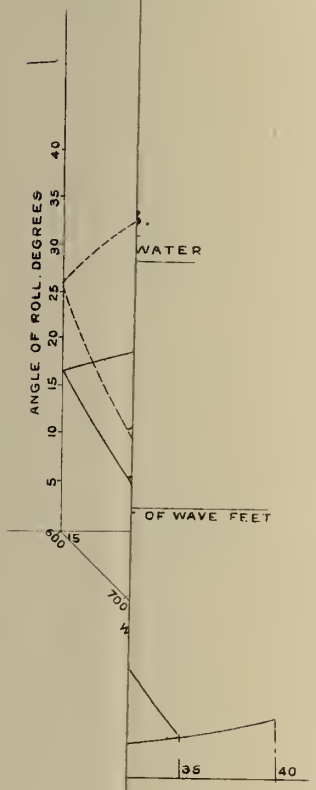


Fig. 22.

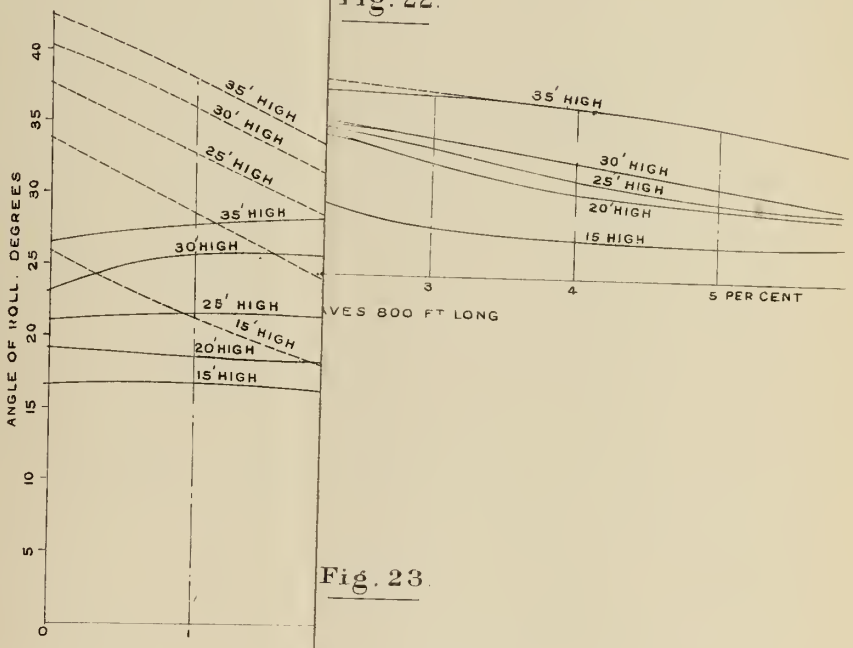


Fig. 23.

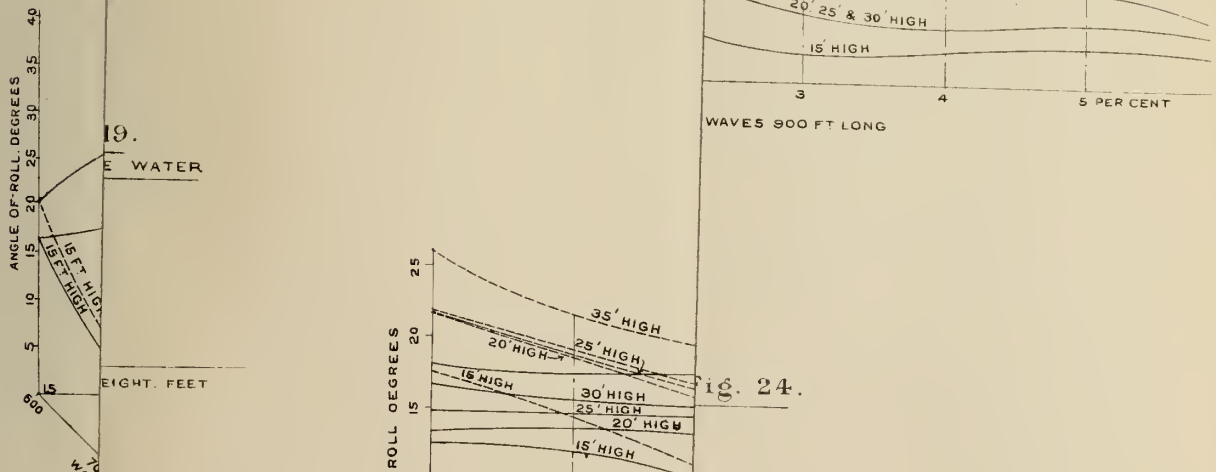
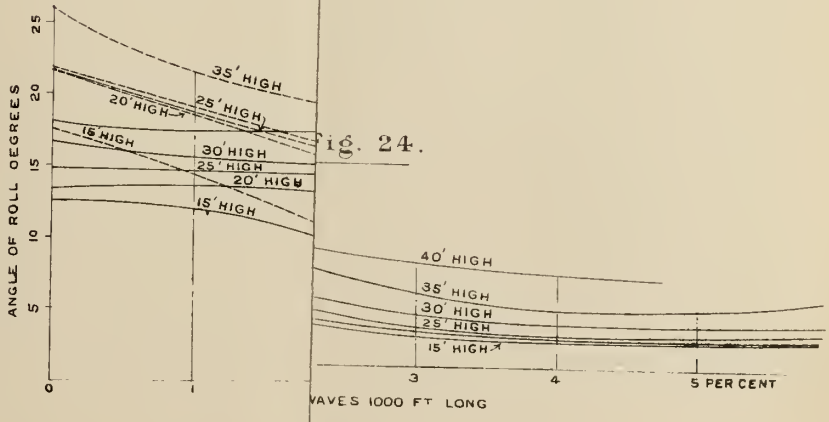


Fig. 24.



To Illustrate Mr. A. Cannon's Paper on "Experimental Determination of the Effect of Internal Loose Water upon the Rolling of a Ship amongst a Regular Series of Waves."

ANGLES OF ROLL FROM NORMAL TO WAVE SURFACE, FOR VARYING WAVE HEIGHTS AND LENGTHS, AND VARYING PERCENTAGES OF FREE WATER IN DOUBLE BOTTOM.

MAXIMUM ANGLES REACHED ARE INDICATED BY DOTTED LINES --------
FORCED OSCILLATIONS " " " " FULL " ————

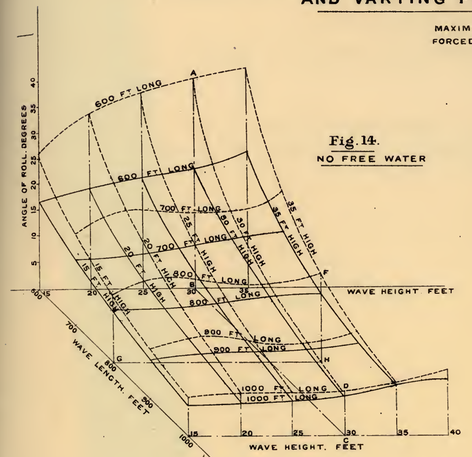


Fig. 14.
NO FREE WATER

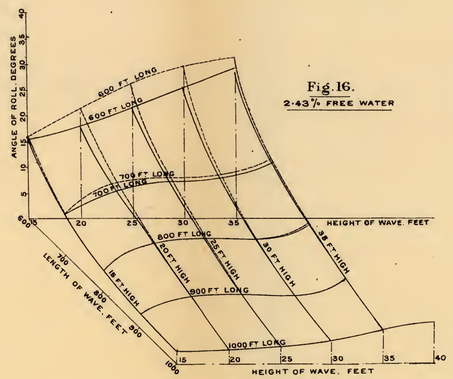


Fig. 16.
2·43% FREE WATER

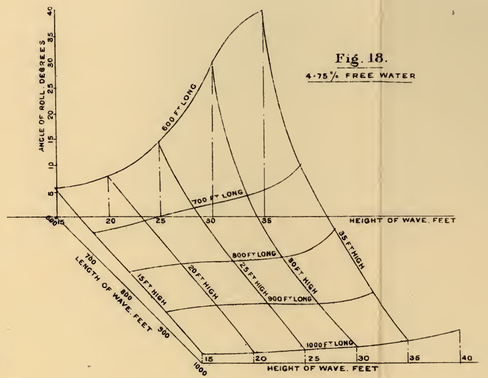


Fig. 18.
4·75% FREE WATER

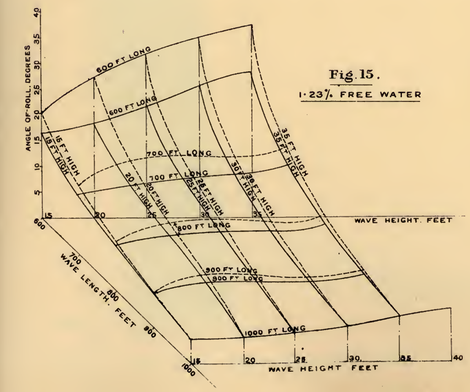


Fig. 15.
1·23% FREE WATER

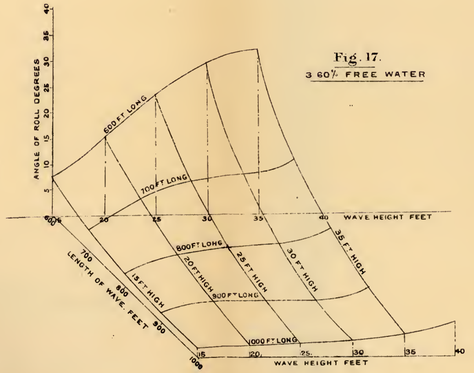


Fig. 17.
3·60% FREE WATER

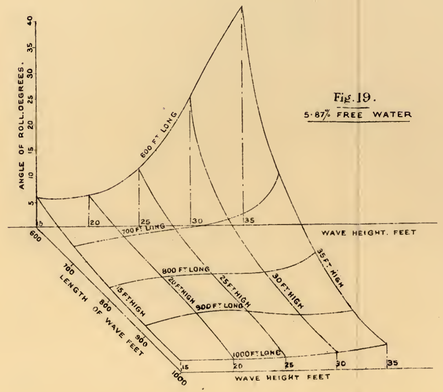


Fig. 19.
5·87% FREE WATER

CURVES OF ANGLES OF ROLL FROM NORMAL TO WAVE SURFACE

EACH CURVE IS FOR A CONSTANT WAVE HEIGHT AND LENGTH AND FOR VARYING QUANTITIES OF LOOSE WATER IN DOUBLE BOTTOM EXPRESSED AS A PERCENTAGE OF THE TOTAL DISPLACEMENT

MAXIMUM ANGLES REACHED ARE INDICATED BY DOTTED LINES ----------
FORCED OSCILLATIONS " " " " FULL " ————

Fig. 20.

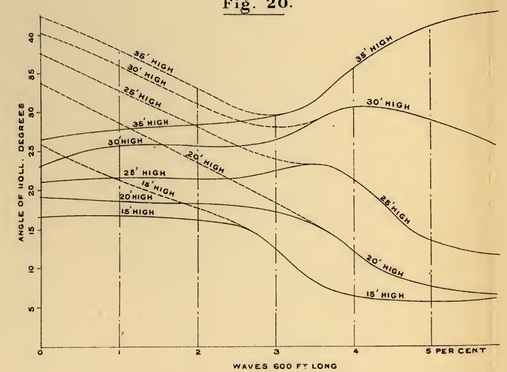


Fig. 22.

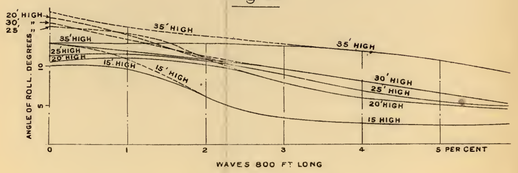


Fig. 21.

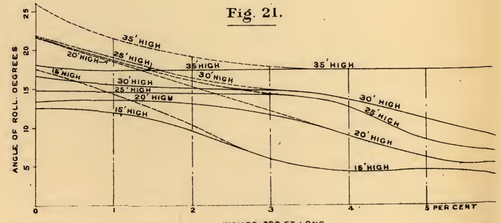


Fig. 23.

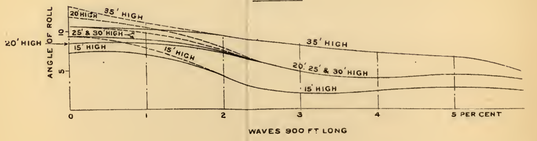


Fig. 24.

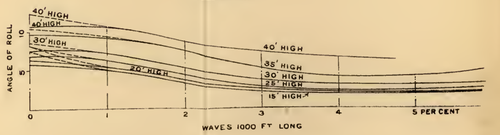


Fig. 3.

MUM ANGLES Φ OF FORCED OSCILLATION IN TANK.

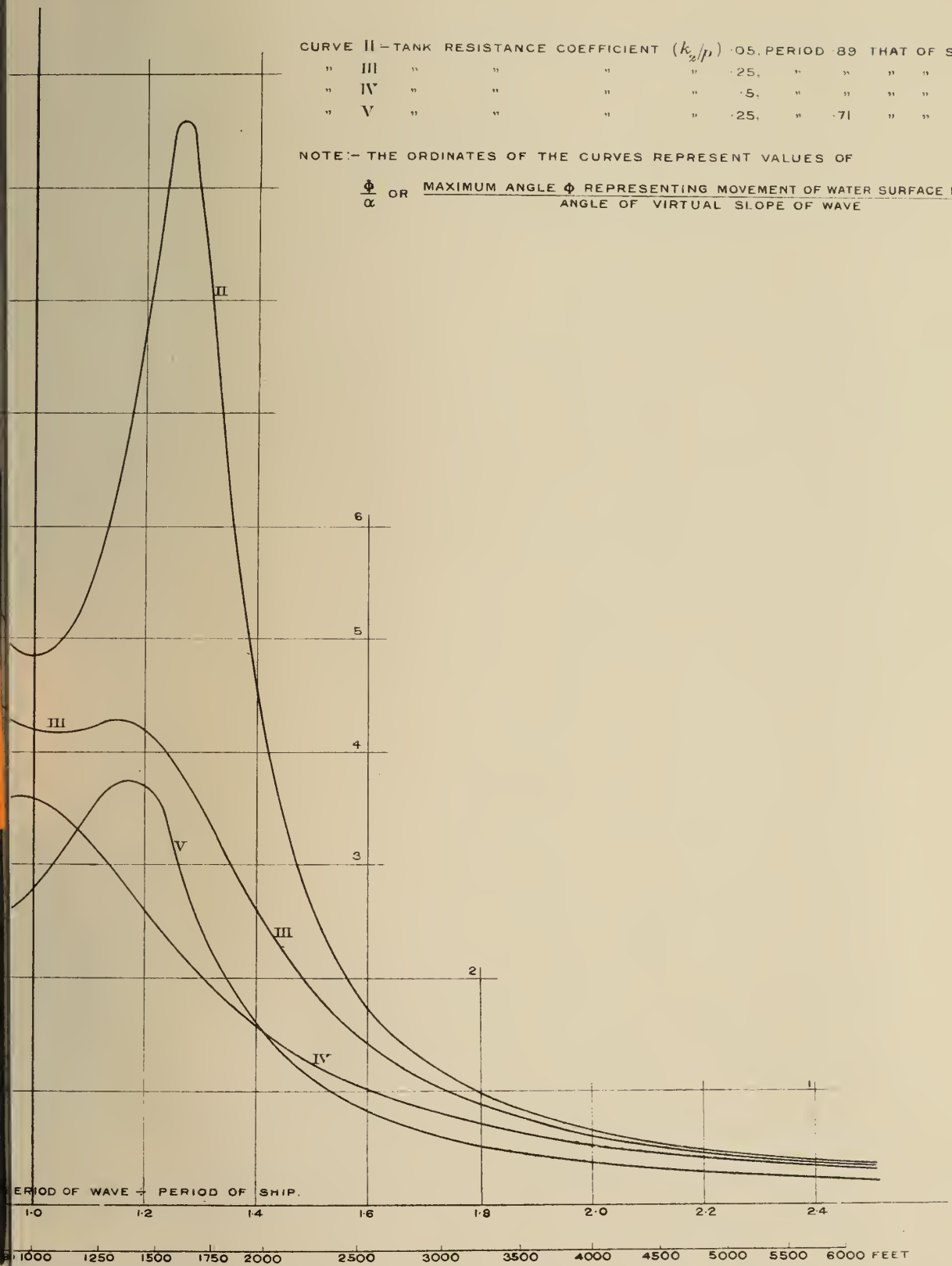


Fig 2.

MAXIMUM ANGLES Θ OF FORCED OSCILLATION OF SHIP.

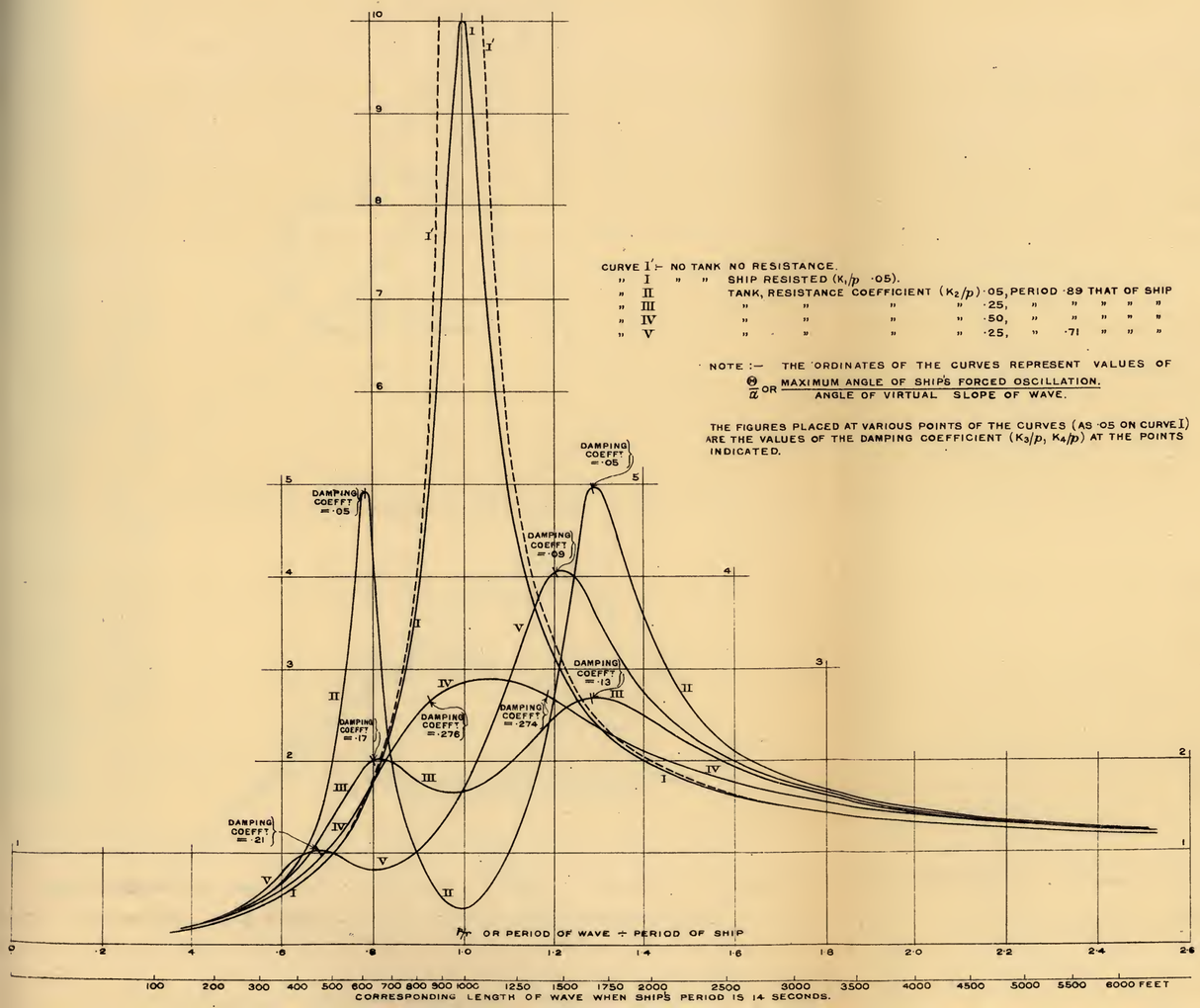

CURVE I' – NO TANK NO RESISTANCE.
" I " " SHIP RESISTED (K_1/p ·05).
" II TANK, RESISTANCE COEFFICIENT (K_2/p) ·05, PERIOD ·89 THAT OF SHIP
" III " " " " ·25, " " " " "
" IV " " " " ·50, " " " " "
" V " " " " ·25, " ·71 " " "
NOTE :– THE ORDINATES OF THE CURVES REPRESENT VALUES OF
Θ/α OR MAXIMUM ANGLE OF SHIP'S FORCED OSCILLATION. / ANGLE OF VIRTUAL SLOPE OF WAVE.
THE FIGURES PLACED AT VARIOUS POINTS OF THE CURVES (AS ·05 ON CURVE I) ARE THE VALUES OF THE DAMPING COEFFICIENT (K_3/p, K_4/p) AT THE POINTS INDICATED.
DAMPING COEFFT = ·05
DAMPING COEFFT = ·05
DAMPING COEFFT = ·09
DAMPING COEFFT = ·13
DAMPING COEFFT = ·17
DAMPING COEFFT = ·276
DAMPING COEFFT = ·274
DAMPING COEFFT = ·21
P/p OR PERIOD OF WAVE ÷ PERIOD OF SHIP
CORRESPONDING LENGTH OF WAVE WHEN SHIP'S PERIOD IS 14 SECONDS.
FEET

Fig. 3.

MAXIMUM ANGLES Φ OF FORCED OSCILLATION IN TANK.

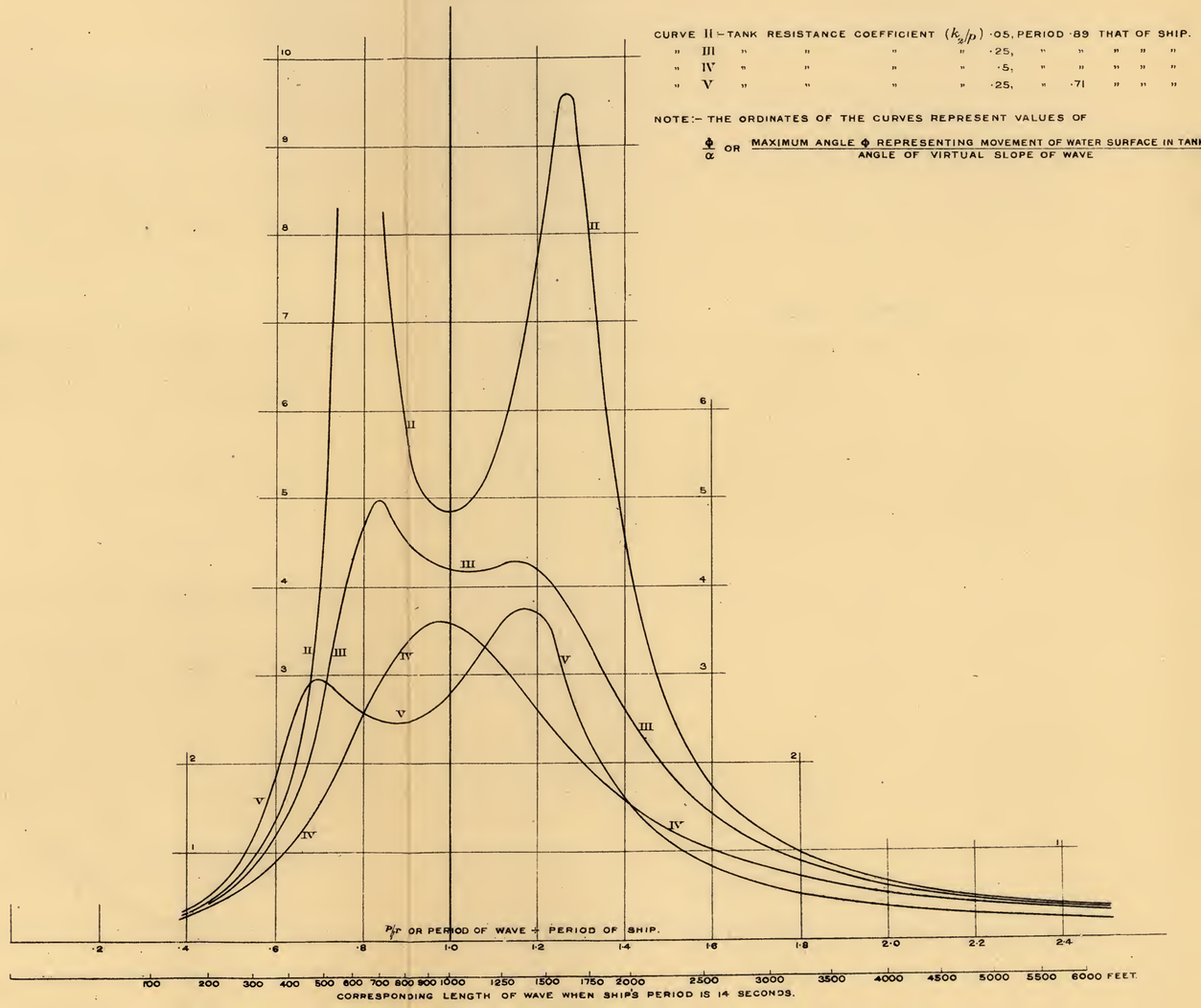

CURVE II – TANK RESISTANCE COEFFICIENT (k_2/p) ·05, PERIOD ·89 THAT OF SHIP.
" III " " " " ·25, " " " " "
" IV " " " " ·5, " " " " "
" V " " " " ·25, " ·71 " " "
NOTE :– THE ORDINATES OF THE CURVES REPRESENT VALUES OF
Φ/α OR MAXIMUM ANGLE Φ REPRESENTING MOVEMENT OF WATER SURFACE IN TANK / ANGLE OF VIRTUAL SLOPE OF WAVE
P/p OR PERIOD OF WAVE ÷ PERIOD OF SHIP.
CORRESPONDING LENGTH OF WAVE WHEN SHIP'S PERIOD IS 14 SECONDS.
FEET

Fig. 4.

TAYLOR'S METHOD OF REPRESENTING
RESULTS OF EXPERIMENTS.

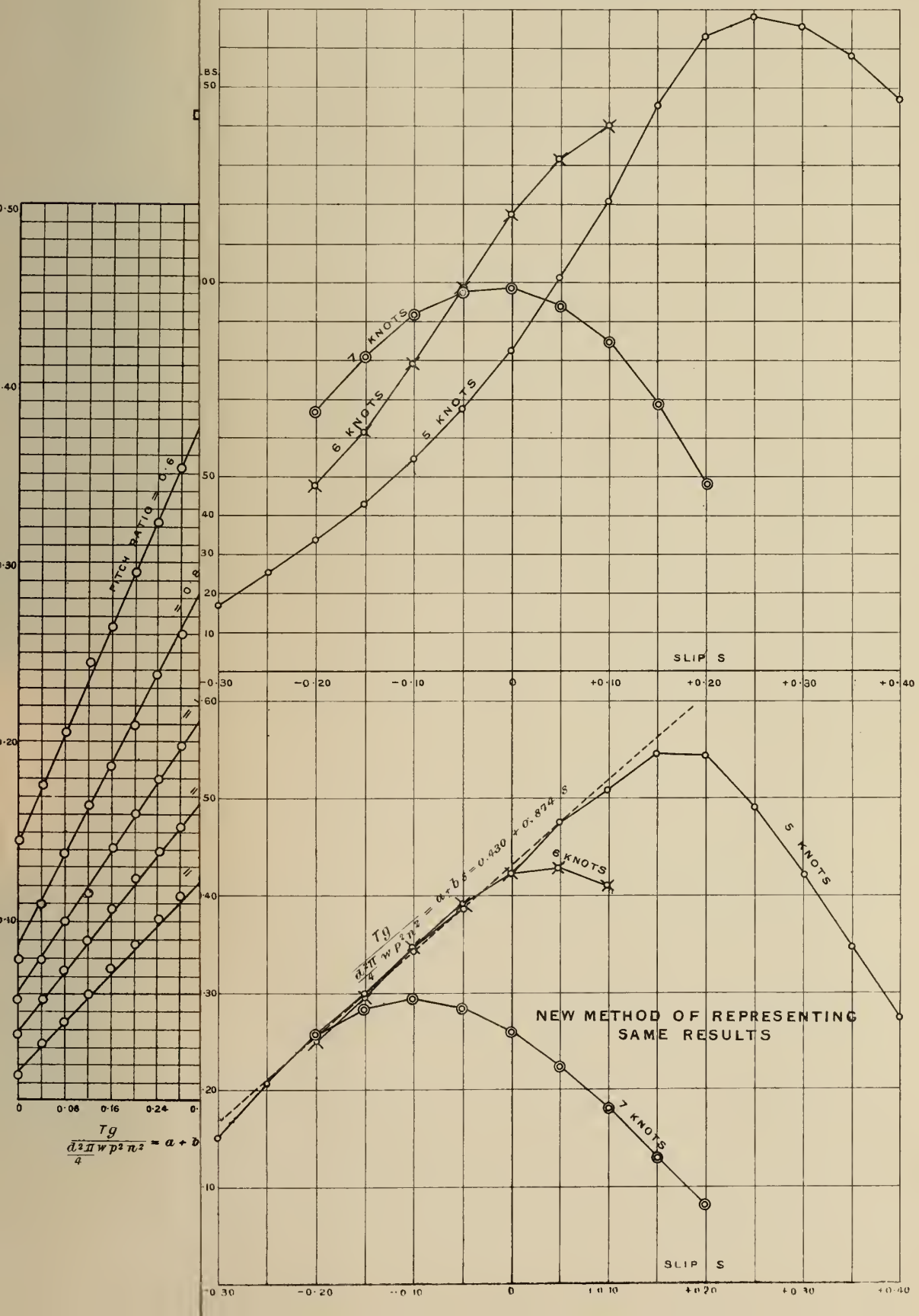


Fig. 5.

D. W. TAYLOR 1904.

$\frac{\text{HELICOIDAL AREA}}{\text{DISC AREA}} = 0.322$

NUMBER OF BLADES = 3

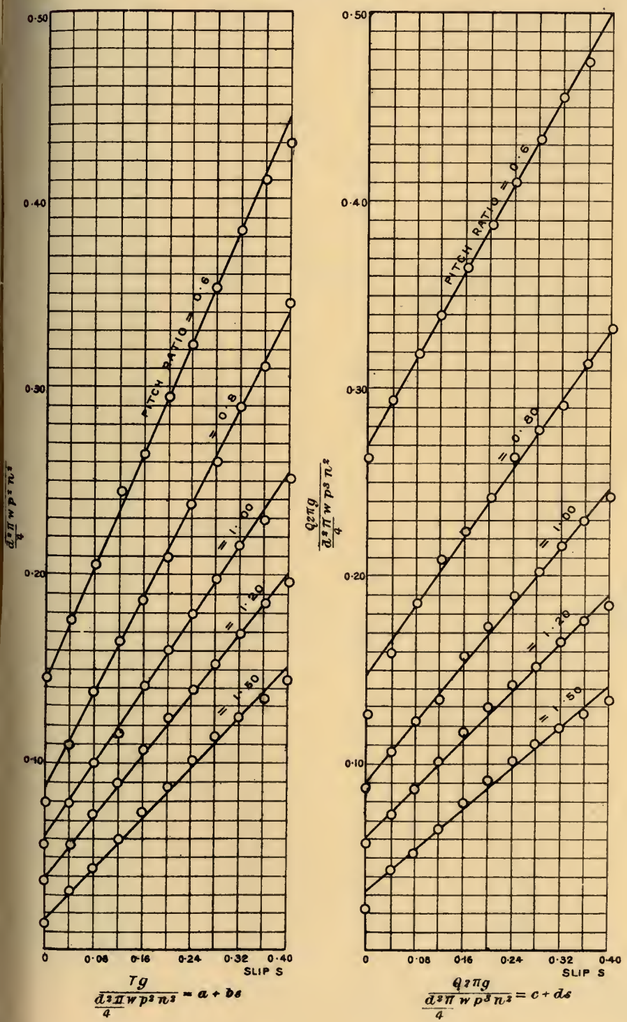


Fig. 3.

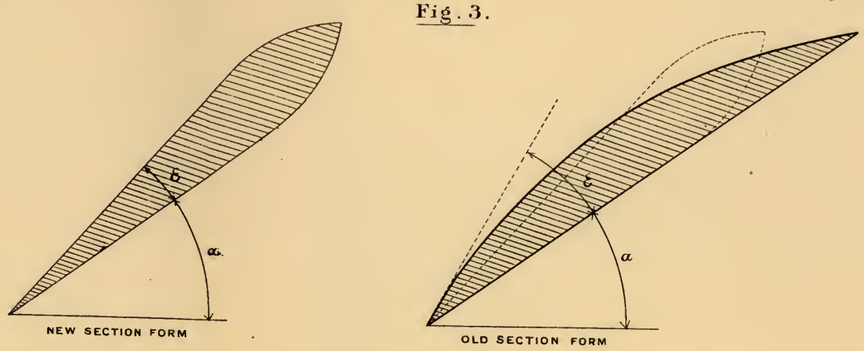


Fig. 6.

TWO SHAFT TURBINE DRIVEN DESTROYER
740 TONS DISPLACEMENT

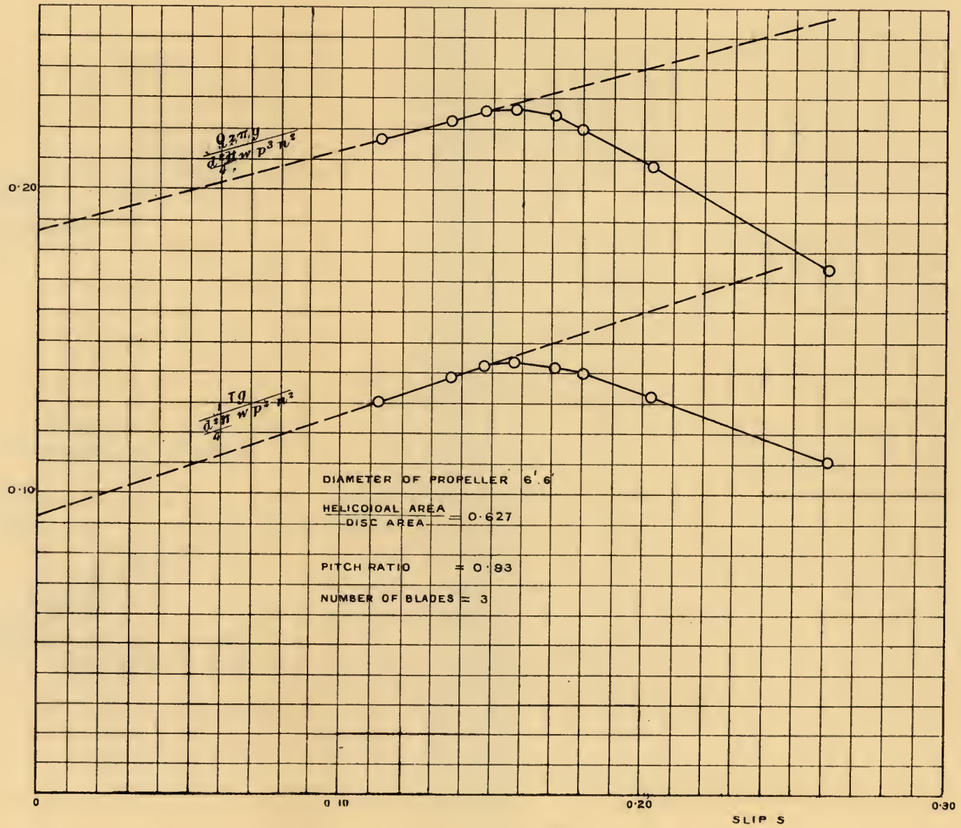
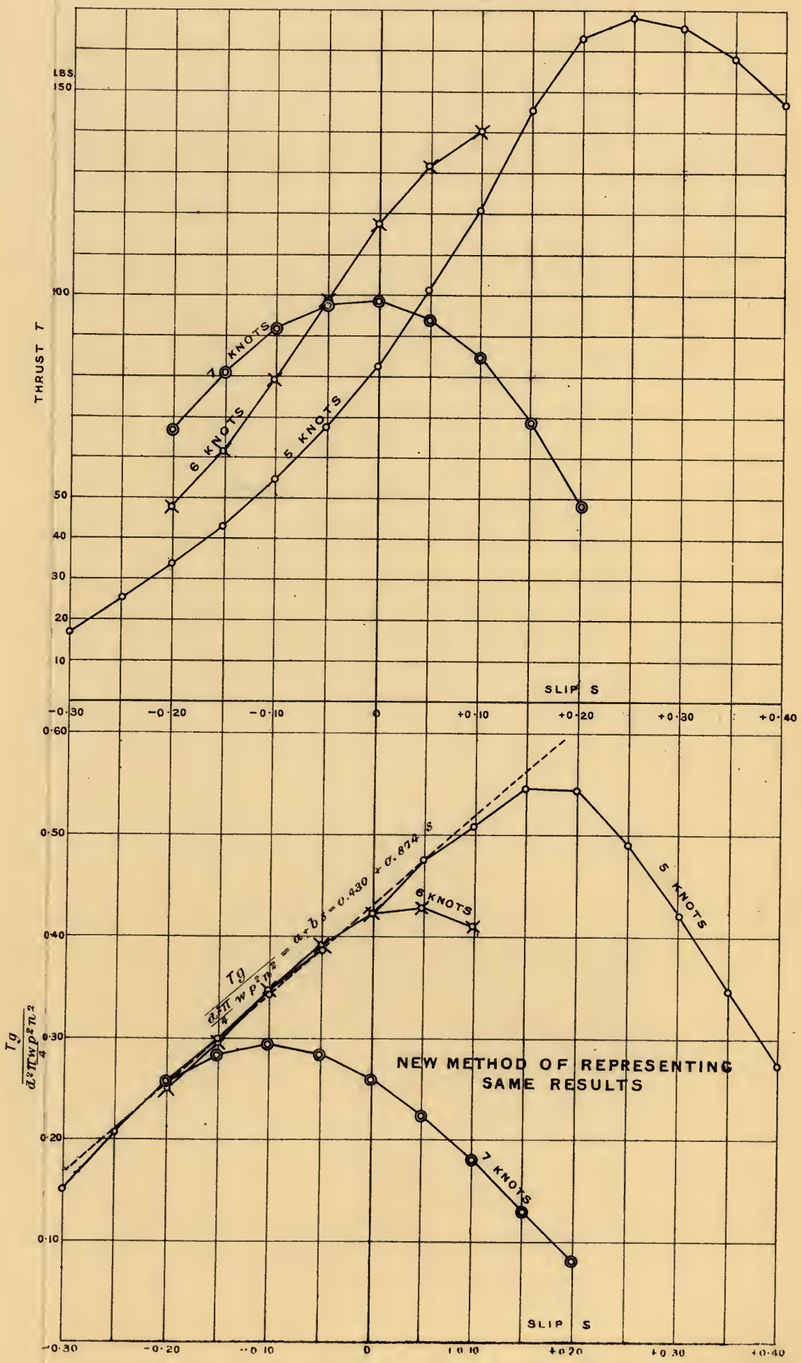


Fig. 4.

TAYLOR'S METHOD OF REPRESENTING RESULTS OF EXPERIMENTS.



To Illustrate Dr. L. Gümbel's Paper, "On the Criterion for the Occurrence of Cavitation."

Fig. 7.

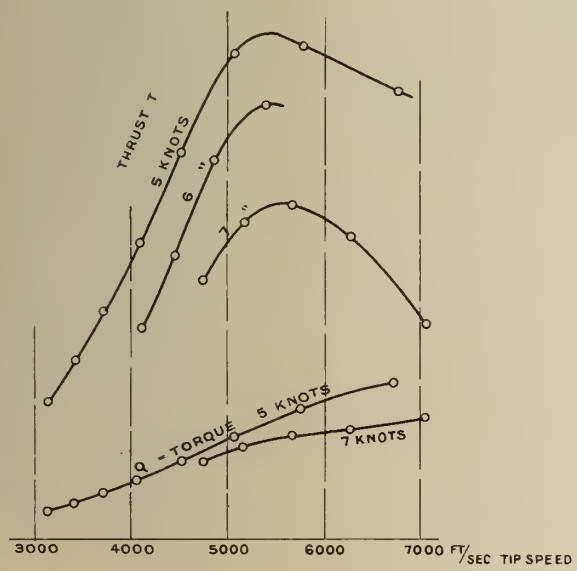


Fig. 8.

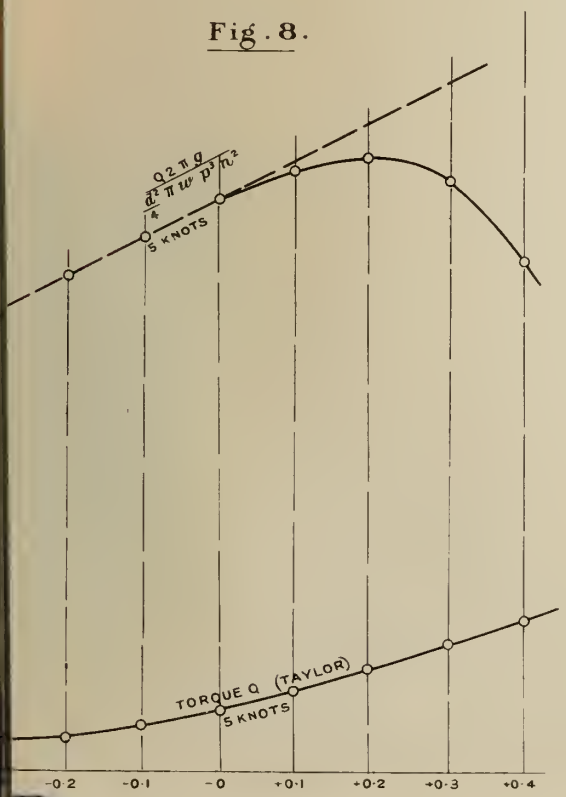


Fig. 9.

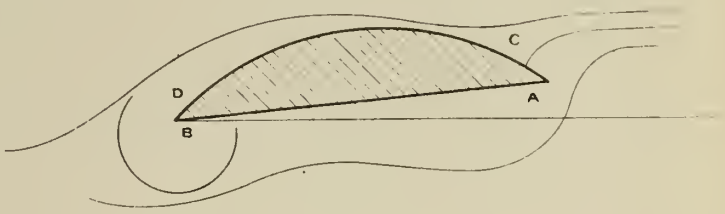


Fig. 10

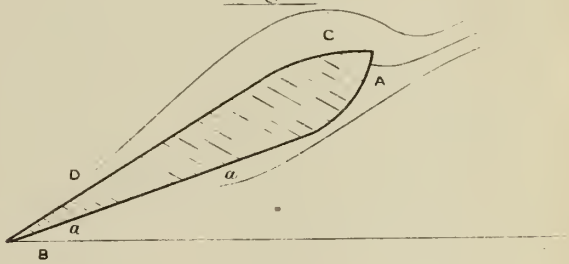


Fig. 11.

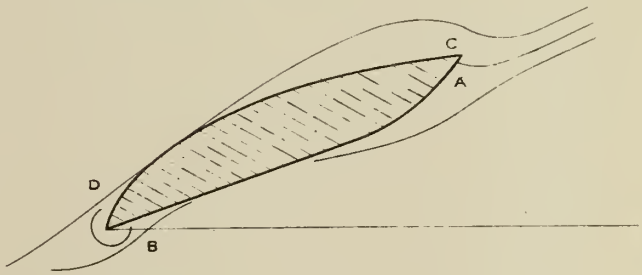


Fig. 12.

CLARK AIR PROPELLER MODEL B

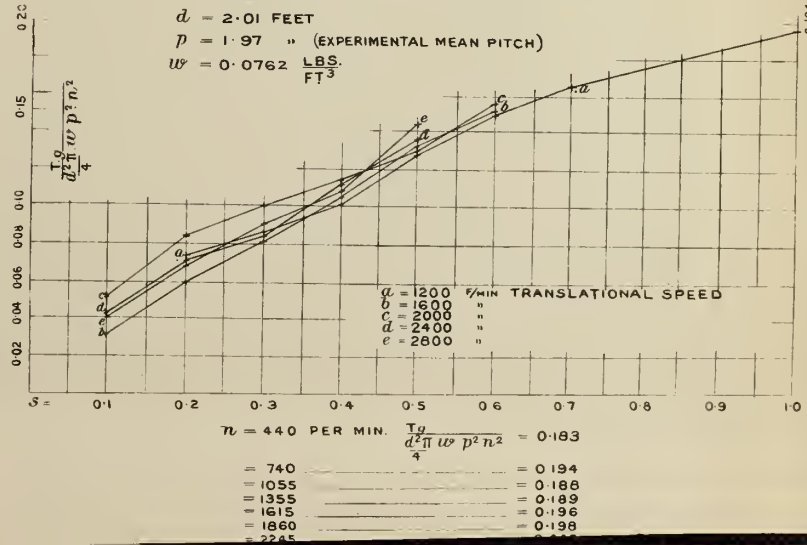


Fig. 3.

PROGRESSIVE TRIAL RESULTS

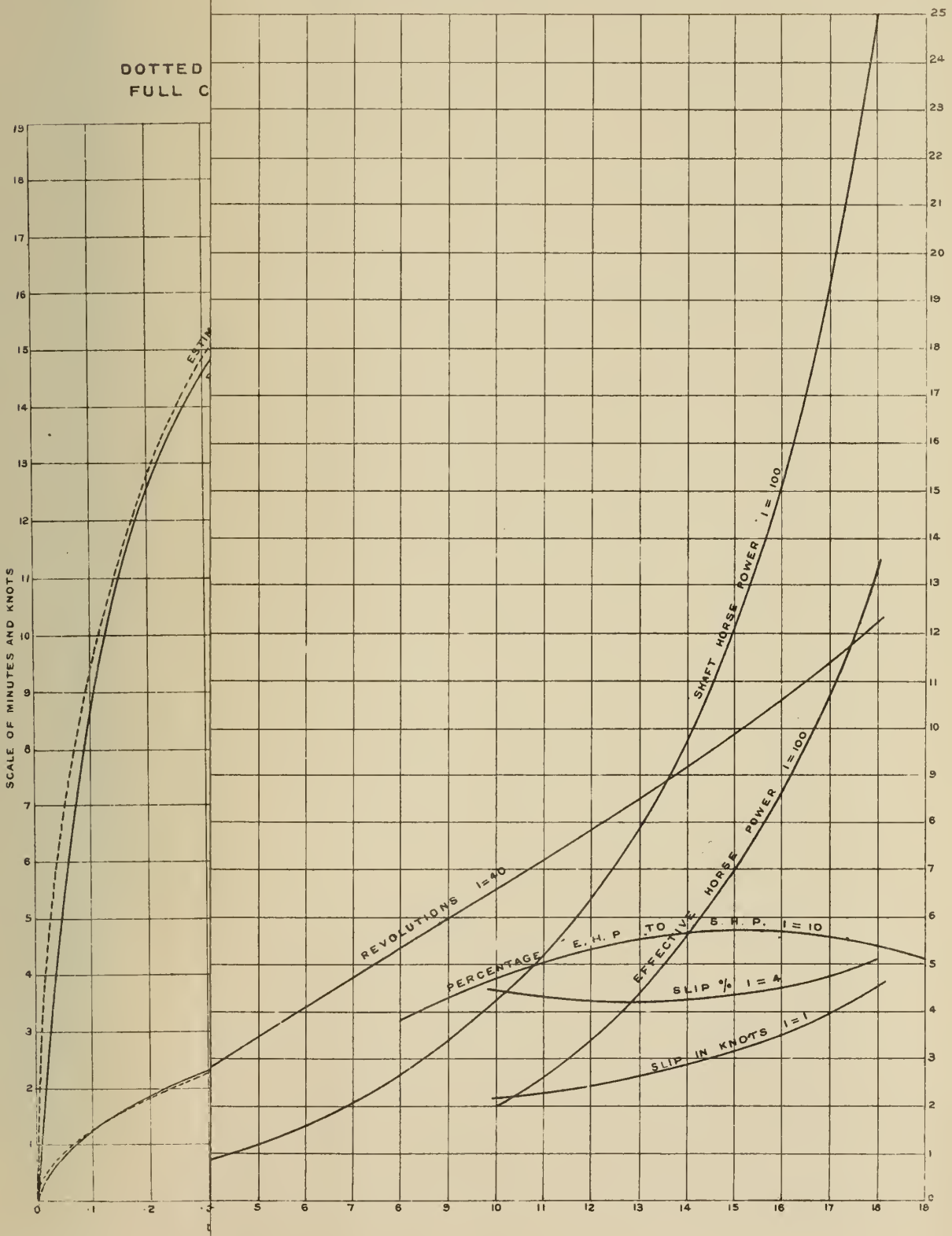

DOTTED
FULL C
ESTIM
SCALE OF MINUTES AND KNOTS
SHAFT HORSE POWER I = 100
REVOLUTIONS I = 40
PERCENTAGE E.H.P TO S.H.P. I = 10
EFFECTIVE HORSE POWER I = 100
SLIP % I = 4
SLIP IN KNOTS I = 1

Fig. 1.

T. S. S. "CURZON"
DOTTED CURVES BY TANK TRIAL
FULL CURVES BY OBSERVATION

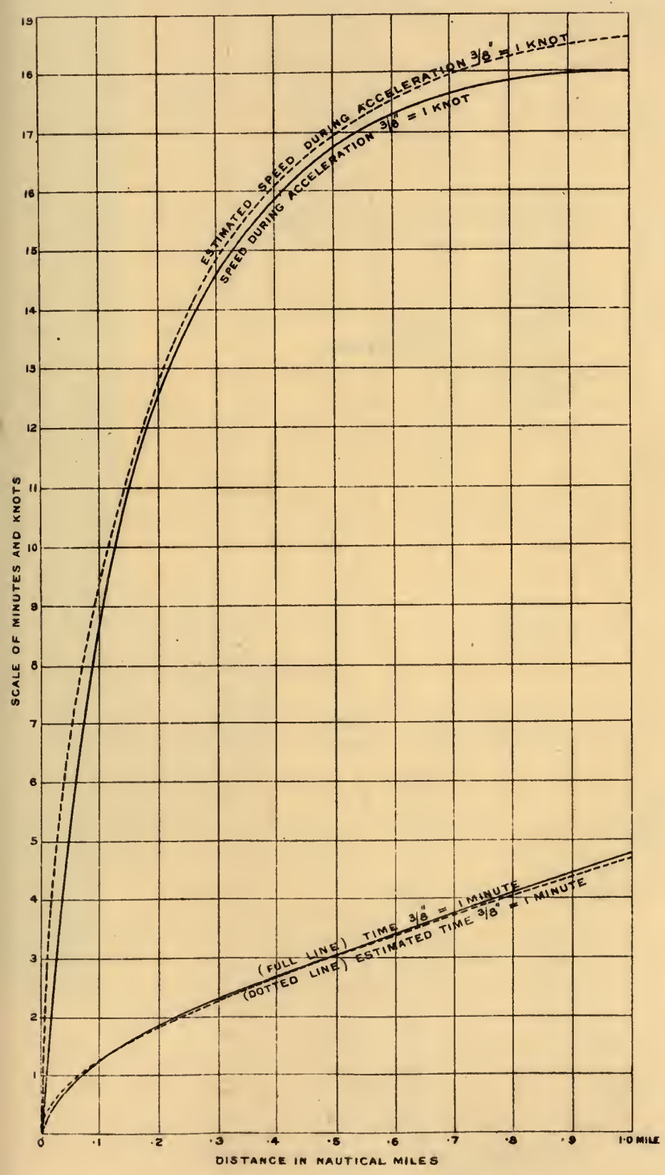

ESTIMATED SPEED DURING ACCELERATION 3/8" = 1 KNOT
SPEED DURING ACCELERATION 3/8" = 1 KNOT
(FULL LINE) TIME 3/8" = 1 MINUTE
(DOTTED LINE) ESTIMATED TIME 3/8" = 1 MINUTE
SCALE OF MINUTES AND KNOTS
DISTANCE IN NAUTICAL MILES
1·0 MILE

Fig. 2.

T. S. S. "CURZON" CURVES ON SPEED BASE.

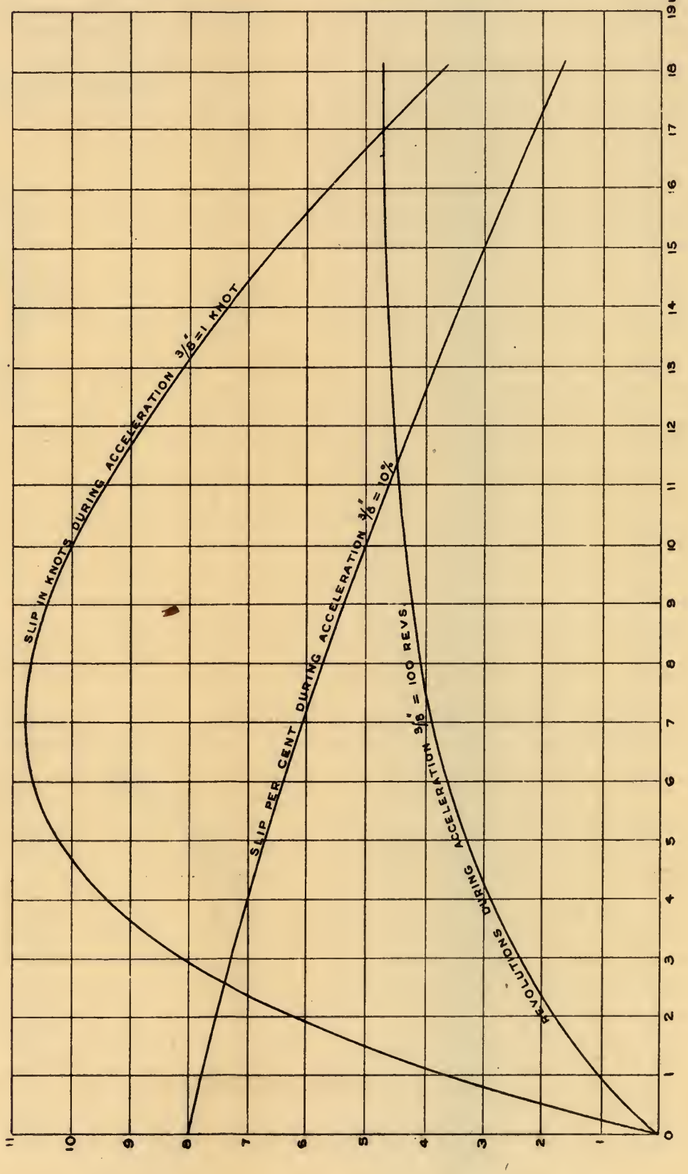
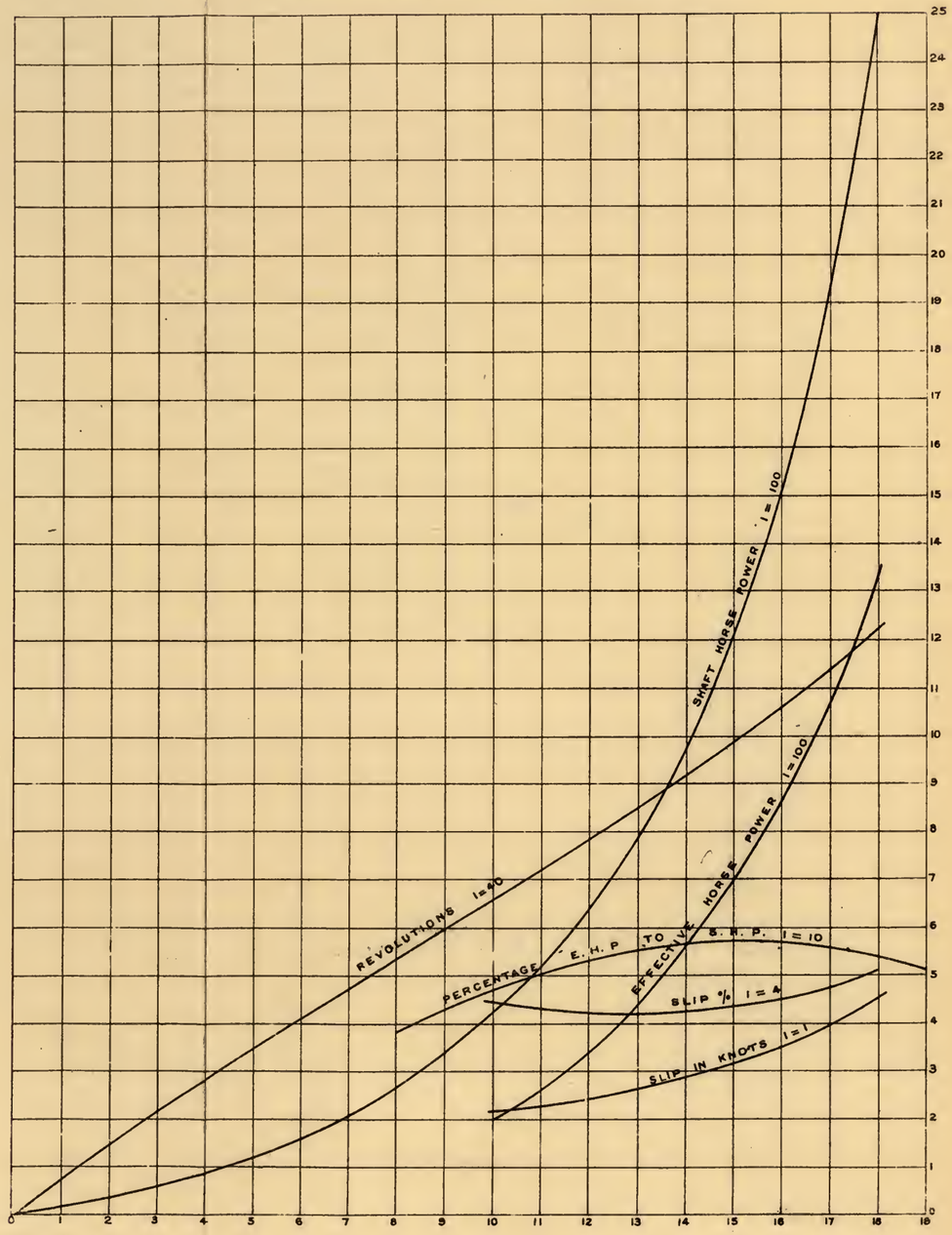

SLIP IN KNOTS DURING ACCELERA
SLIP PER CENT DURING ACCELERATION 3/8" = 10%
REVOLUTIONS DURING ACCELERATION 3/8" = 100 REVS.
19 KNOTS

Fig. 3.

PROGRESSIVE TRIAL RESULTS



urbines" Fig. 3.

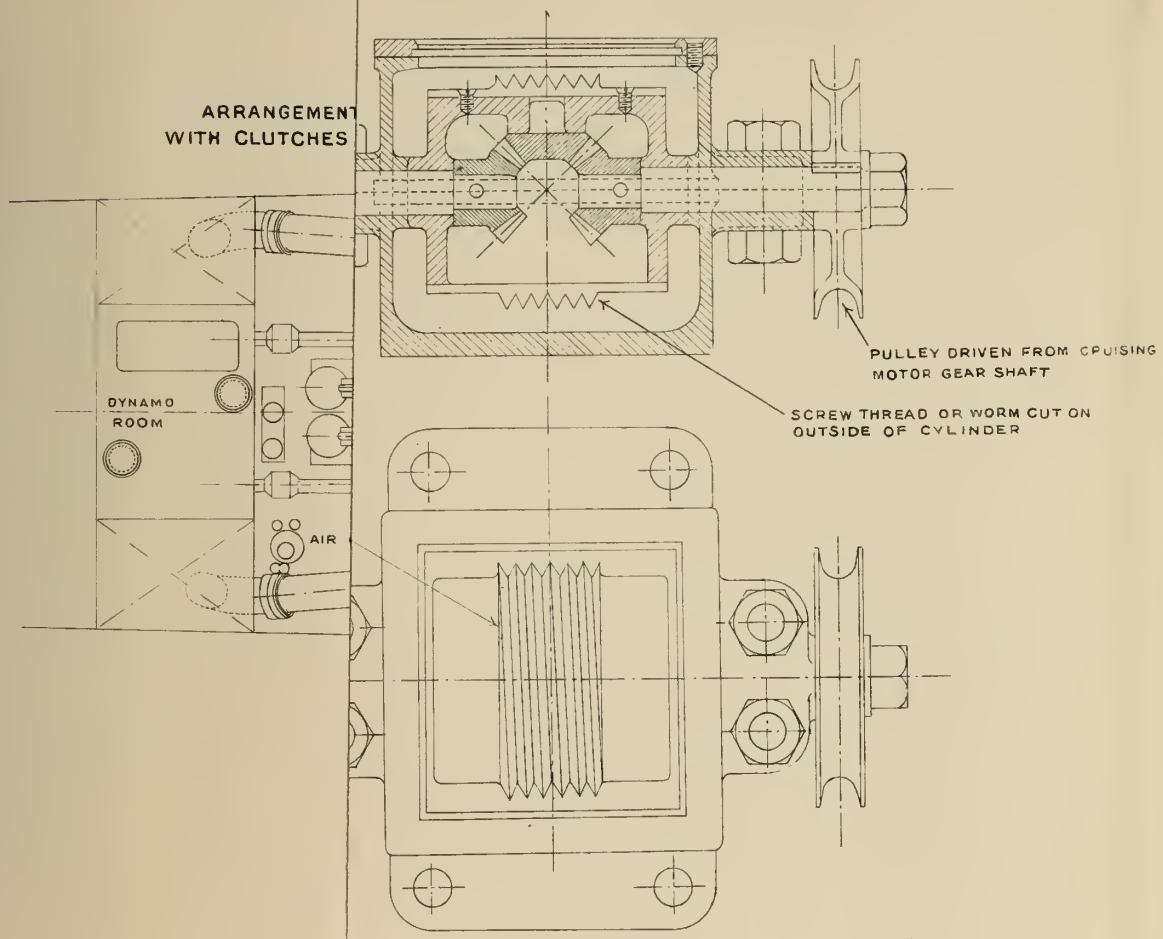
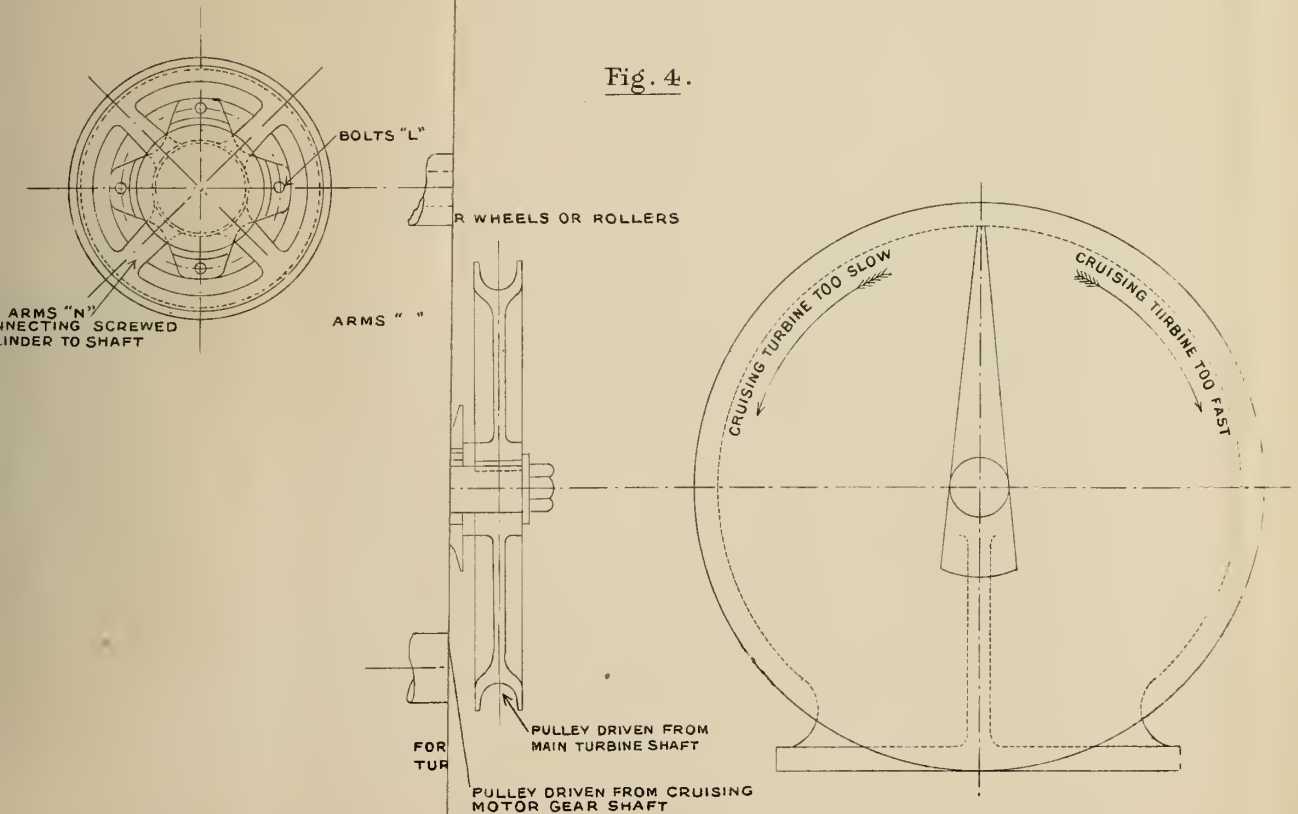


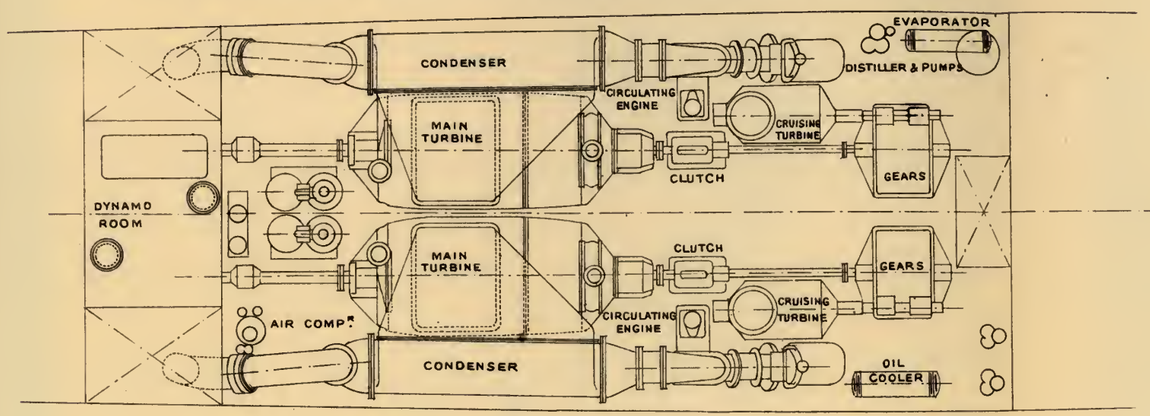
Fig. 4.



To Illustrate Mr. Harold E. Yarrow's Paper on "A Device to Facilitate the Coupling of Cruising Turbines"

Fig. 1.

ARRANGEMENT OF MAIN TURBINES AND GEARED CRUISING TURBINES WITH CLUTCHES FOR DISCONNECTING CRUISING TURBINES FOR FULL SPEED.



ARRANGEMENT OF ENGINE ROOM

Fig. 3.

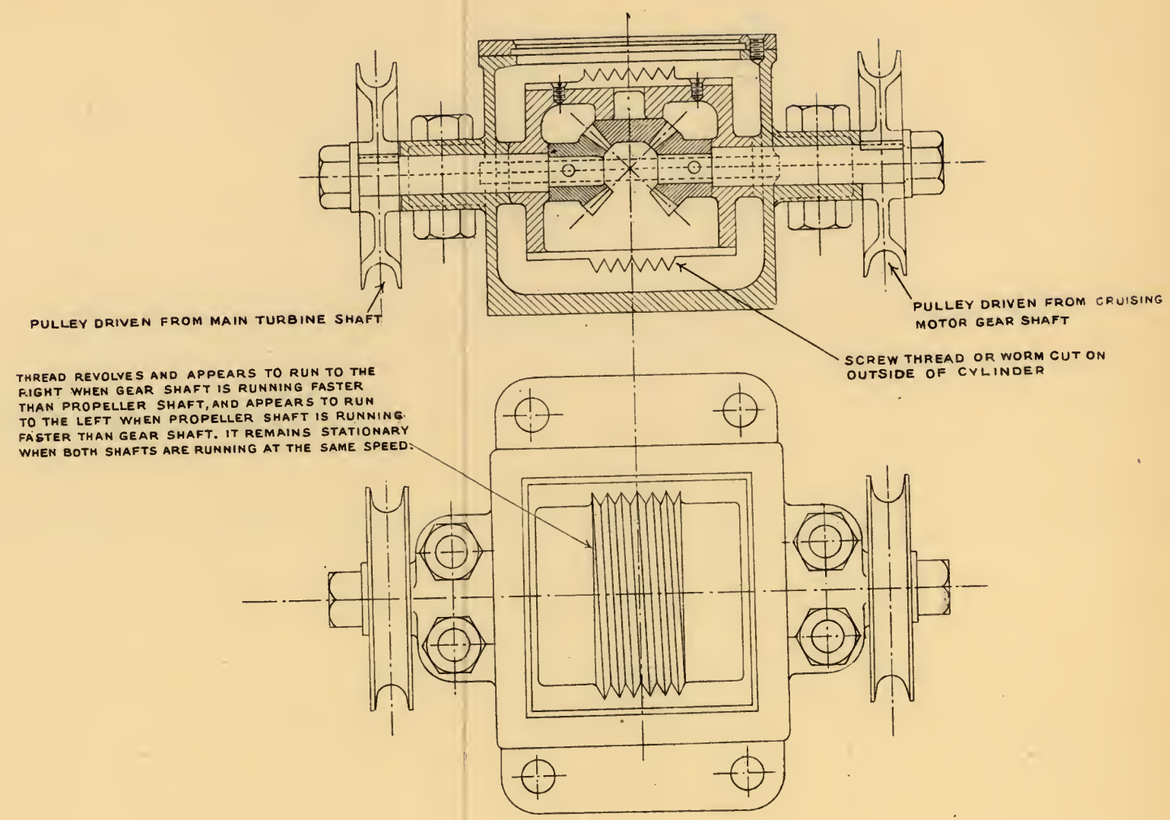
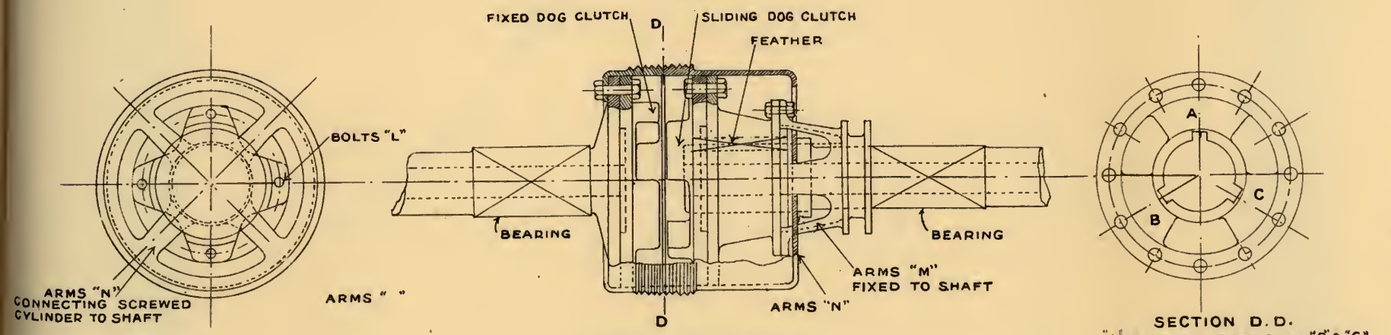
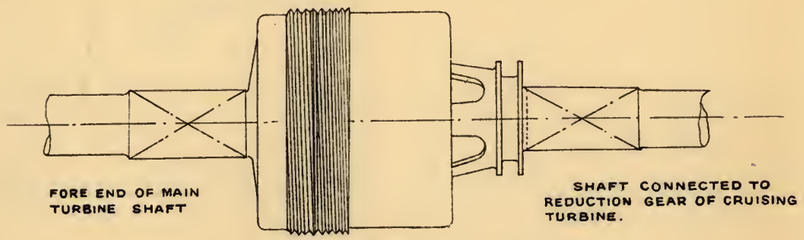

PULLEY DRIVEN FROM MAIN TURBINE SHAFT
PULLEY DRIVEN FROM CRUISING MOTOR GEAR SHAFT
SCREW THREAD OR WORM CUT ON OUTSIDE OF CYLINDER
THREAD REVOLVES AND APPEARS TO RUN TO THE RIGHT WHEN GEAR SHAFT IS RUNNING FASTER THAN PROPELLER SHAFT, AND APPEARS TO RUN TO THE LEFT WHEN PROPELLER SHAFT IS RUNNING FASTER THAN GEAR SHAFT. IT REMAINS STATIONARY WHEN BOTH SHAFTS ARE RUNNING AT THE SAME SPEED.

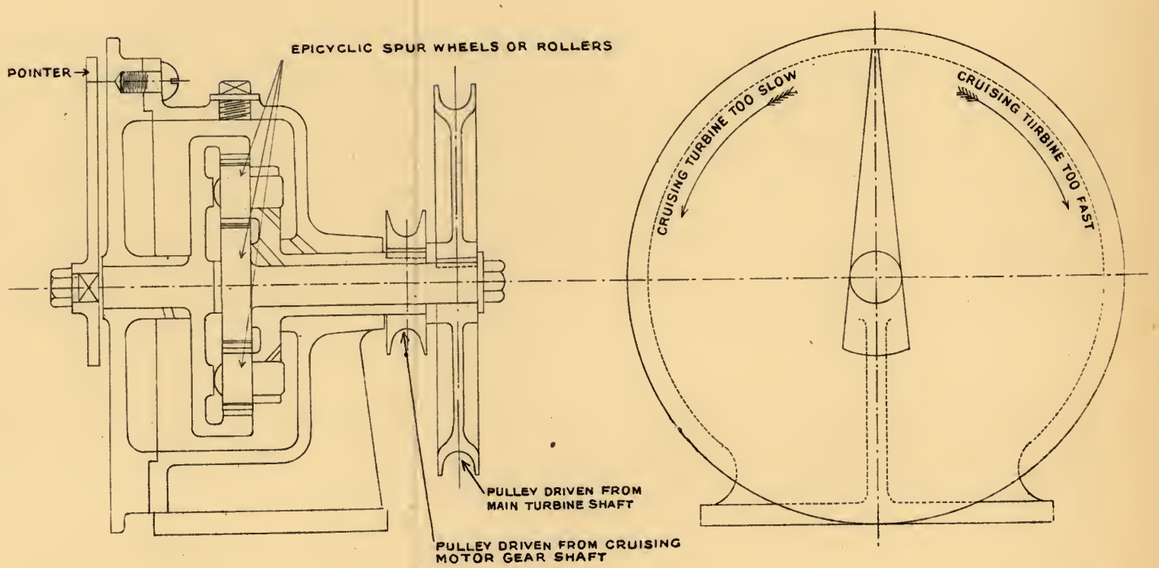
Fig. 2.





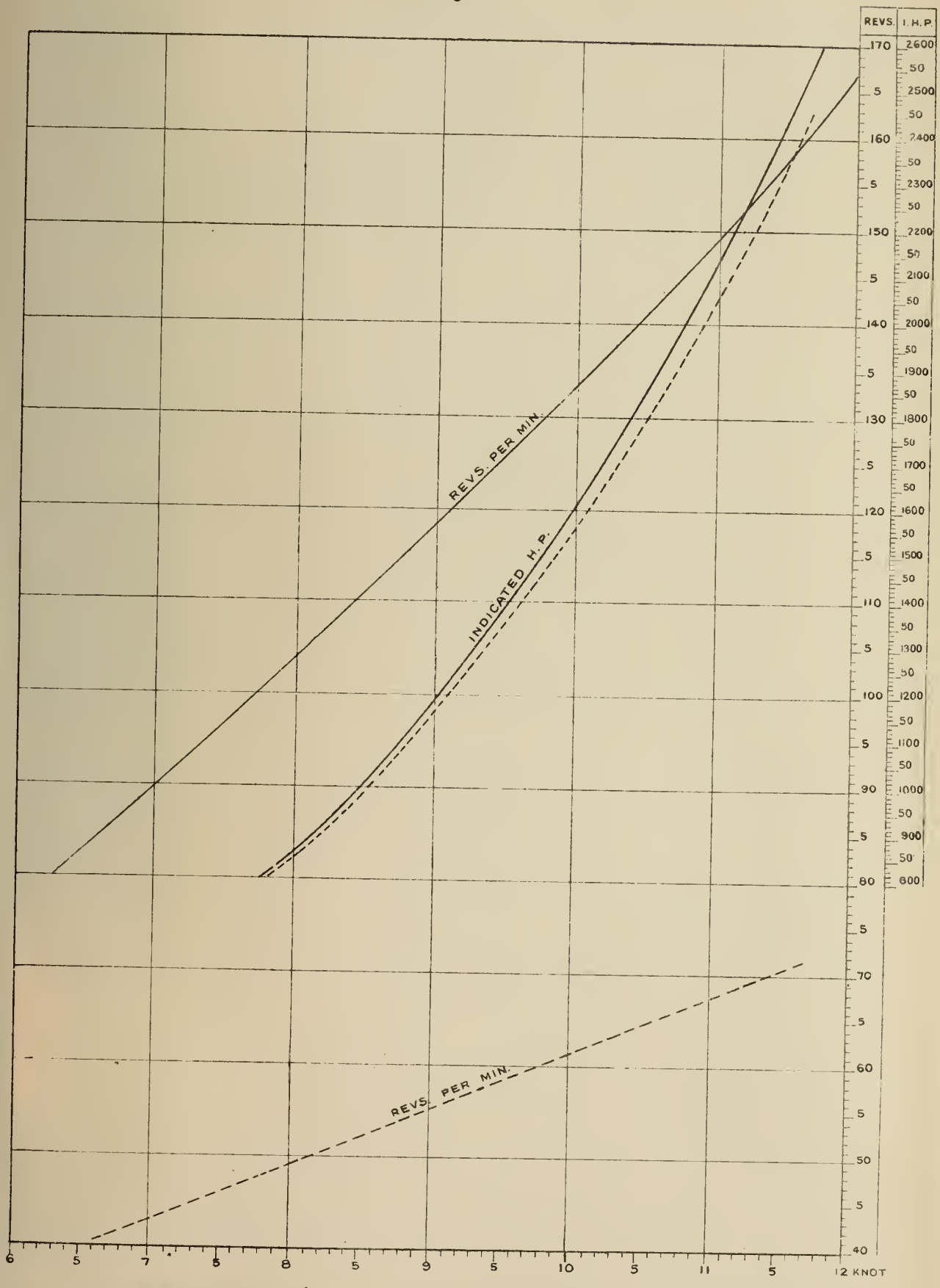
OUTSIDE VIEW OF CLUTCH

Fig. 4.


POINTER
EPICYCLIC SPUR WHEELS OR ROLLERS
PULLEY DRIVEN FROM MAIN TURBINE SHAFT
PULLEY DRIVEN FROM CRUISING MOTOR GEAR SHAFT
CRUISING TURBINE TOO SLOW
CRUISING TURBINE TOO FAST

To Illustrate Mr. I. Knudsen's Paper, "Performance on Service of the Motor Ship 'Suecia.'"

Fig. 1.



M.S. "SUECIA" (FULL LINES)
362′ x 51′.3″ x 23′ (DRAUGHT) D.W. 6500 TONS

S.S. "PRINCESSAN INGEBORG" (DOTTED LINES)
360′ x 48′.9″ x 22′.4″ (DRAUGHT) D.W. 5895 TONS

LIBRARY
OF THE
UNIVERSITY OF

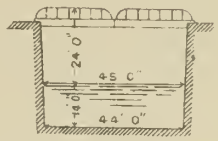
To Illustrate Messrs. John Reid and H. A. Mavor's Paper,

"A Case for Electric Propulsion."

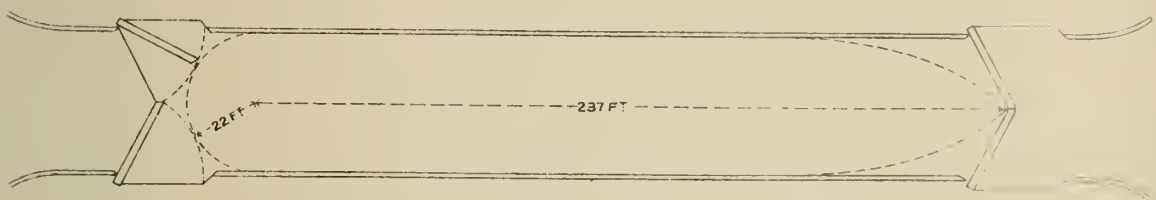
Fig. 1.

WELLAND CANAL.

NOTE: THE SHAPE OF THE STERN OF A VESSEL REGULATES THE EXTREME LENGTH THAT CAN BE LOCKED THROUGH.
A NARROW VESSEL CAN BE LONGER THAN A WIDE ONE
A LONGER VESSEL CAN BE LOCKED DOWN THAN UP.
NO VESSEL SHOULD BE OVER 44 FEET EXTREME WIDTH.

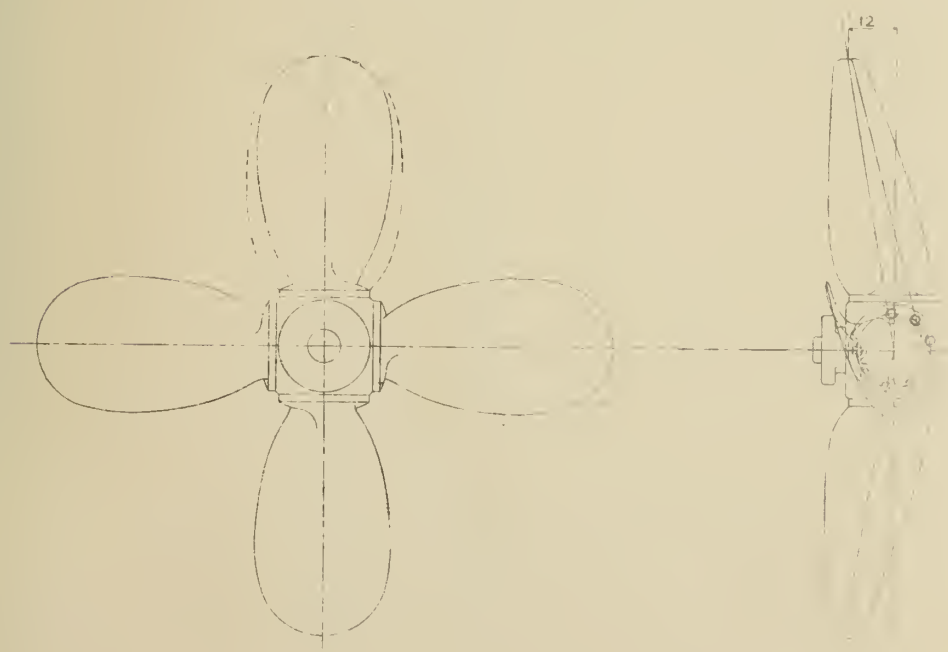


SECTION THROUGH CHAMBER.



PLAN OF LOCK.

Fig. 2.



SCALE 1 IN = 4 FEET

PARTICULARS

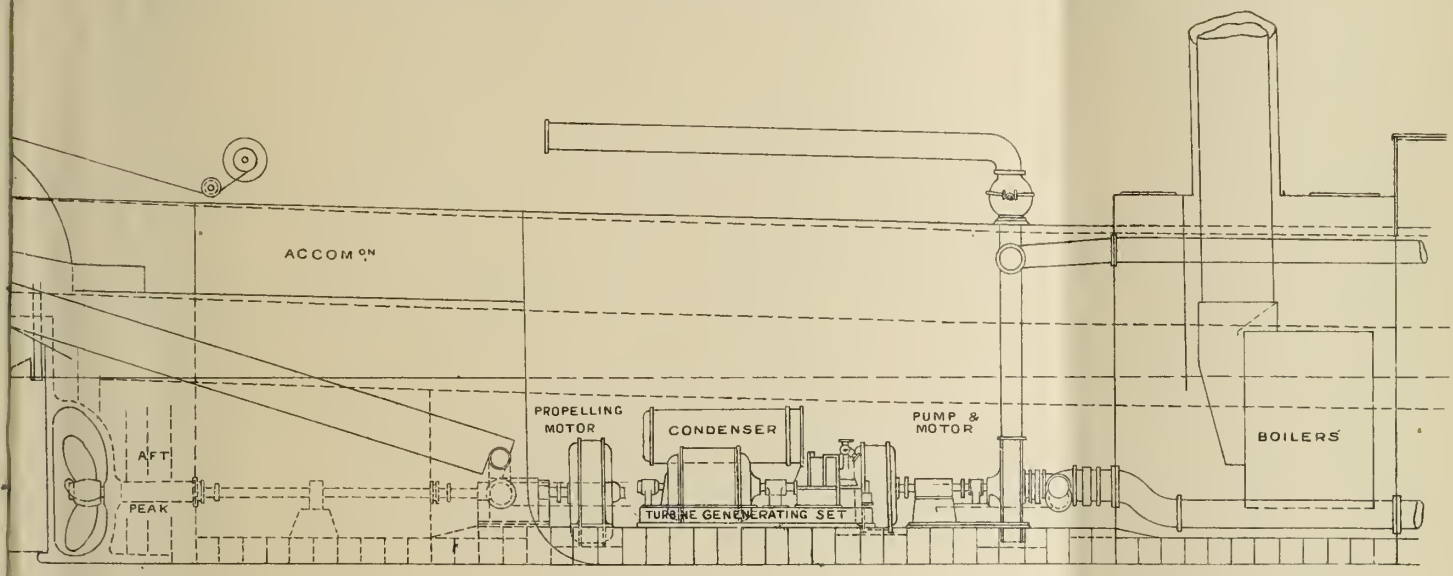
DIAMETER OF PROPELLER	12	0"
PITCH " "	13	0"

SURFACE

PROJECTED	45	SQUARE FEET
DEVELOPED	55	" "

Fig. 6.

DESIGN FOR DREDGER WITH ELECTRIC PROPULSION



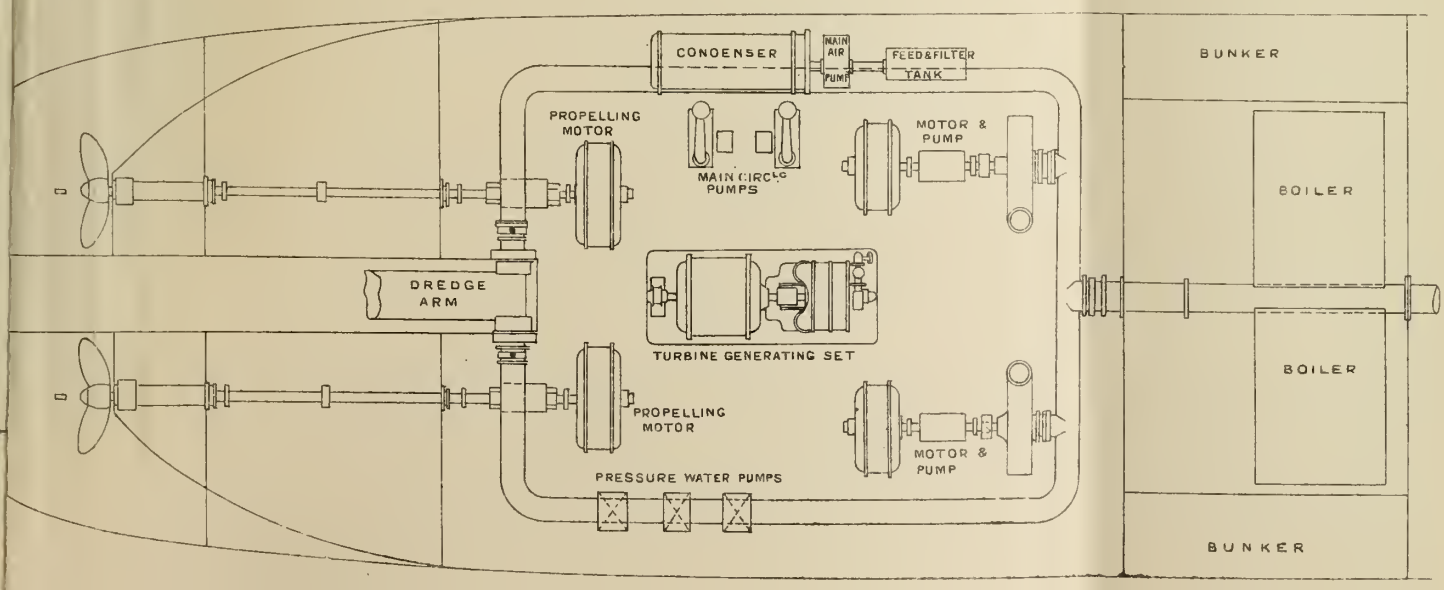
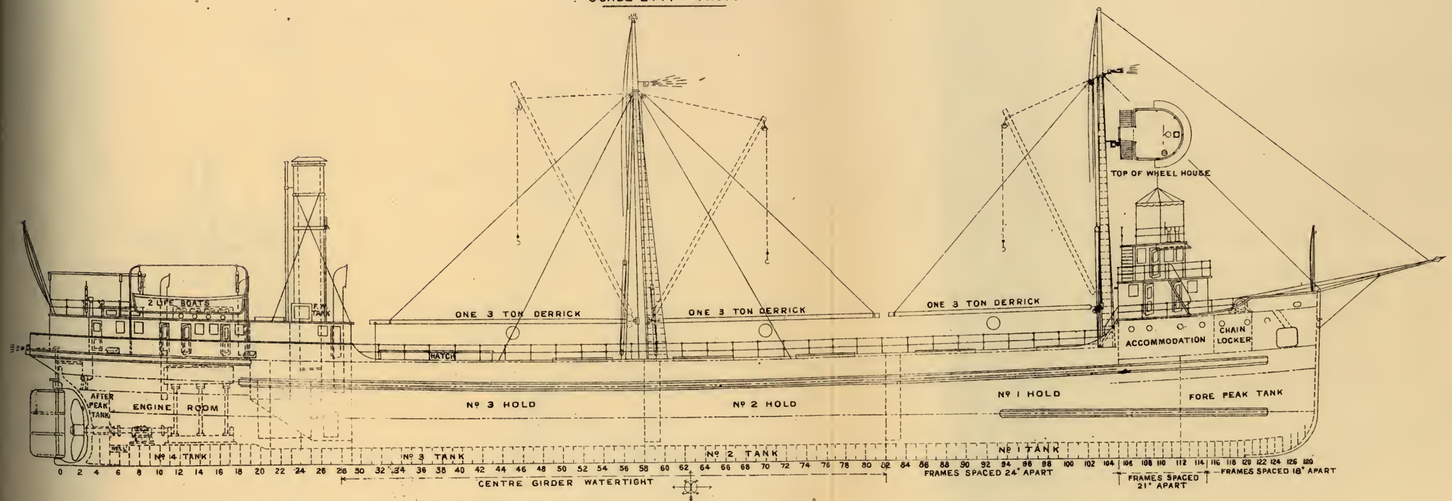


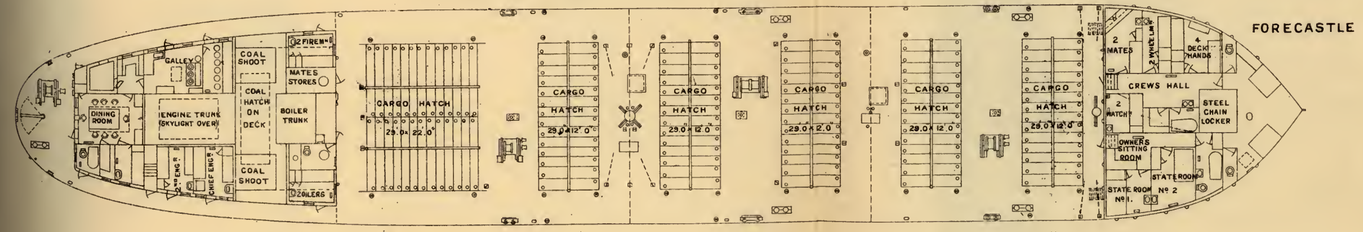
Fig. 3.

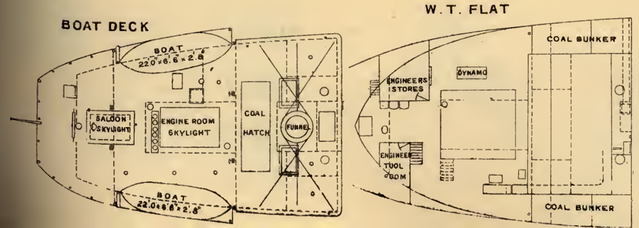
GENERAL ARRANGEMENT OF CANADIAN LAKE STEAMER

DIMENSIONS:- 250.0' B.P. × 42.6' × 20.0' MLD

SCALE 24 FT. = 1 INCH.







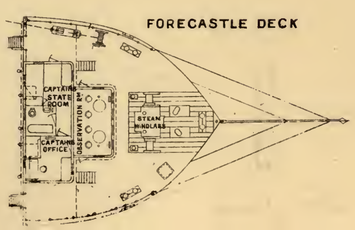
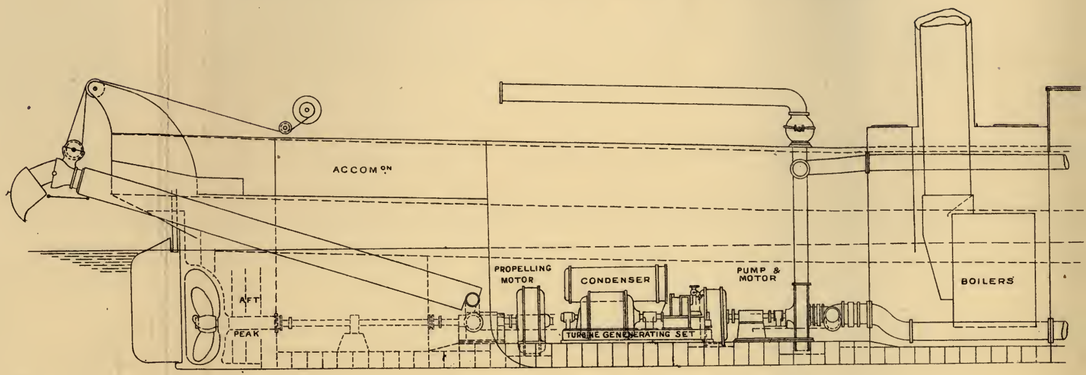


Fig. 6.

DESIGN FOR DREDGER WITH ELECTRIC PROPULSION



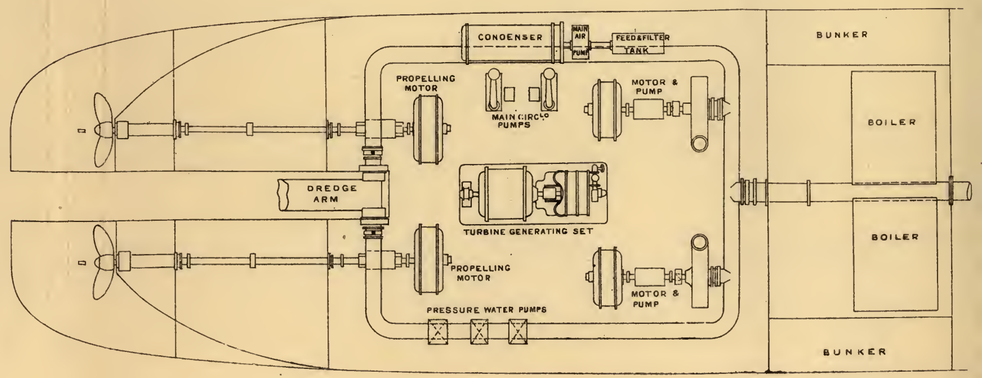
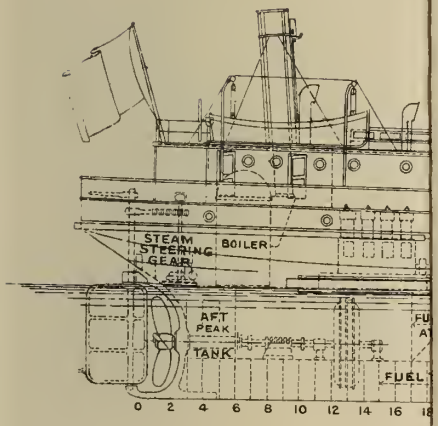
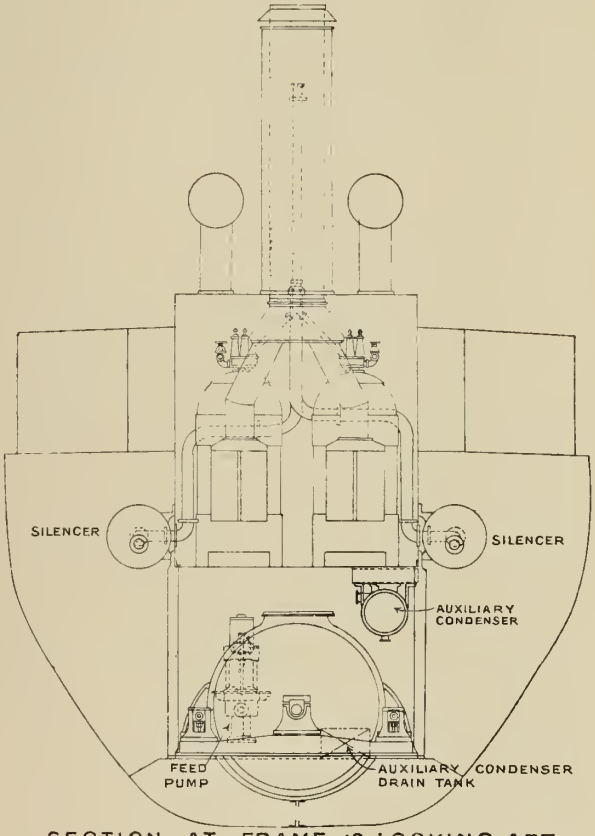


Fig. 4.

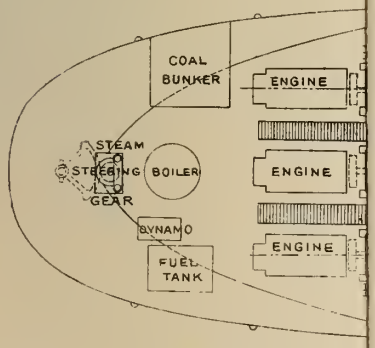
DESIGN FOR LAKE V OPELLED VESSEL.
WITH DIESEL ENGI
ELECTRIC TRANSM

NUTE.

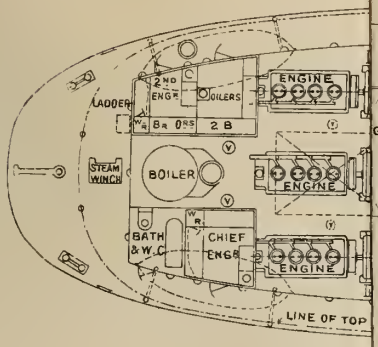




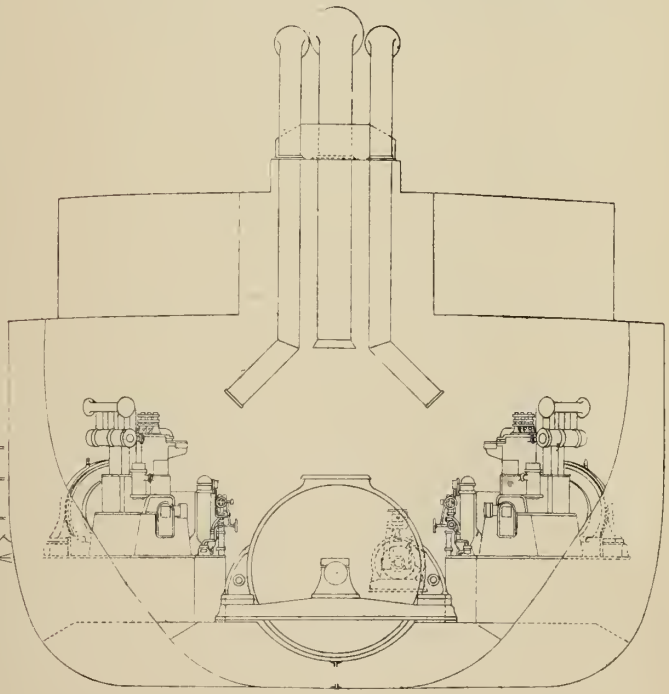
SECTION AT FRAME 13 LOOKING AFT.



LOWER DECK



POOP DECK

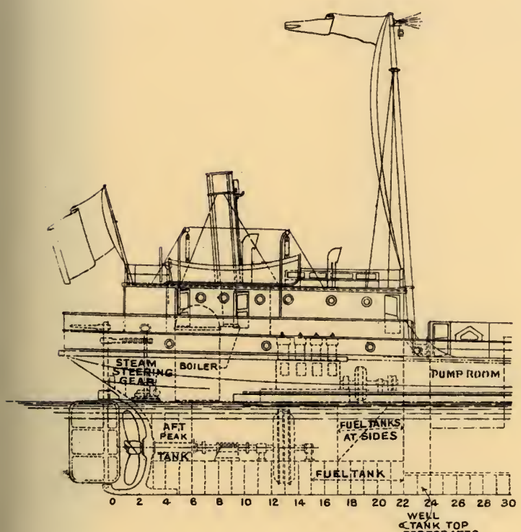


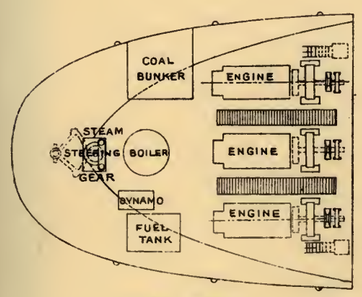
SECTION AT FRAME 13 LOOKING FORWARD.

To Illustrate Messrs. John Reid and H. A. Mavor's Paper, "A Case for Electric Propulsion."

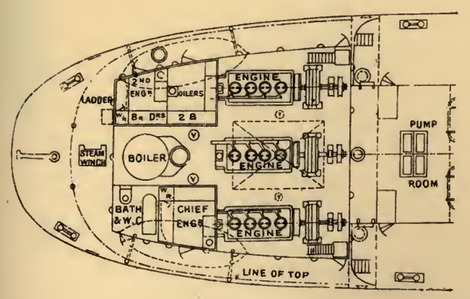
Fig. 4.

DESIGN FOR LAKE VESSEL WITH DIESEL ENGINES & ELECTRIC TRANSMISSION


STEAM STEERING GEAR
BOILER
PUMP ROOM
AFT PEAK TANK
FUEL TANKS AT SIDES
FUEL TANK
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
WELL TANK TOP PERFORATED



LOWER DECK



POOP DECK

Fig. 5.

ARRANGEMENT OF MACHINERY FOR AN ELECTRICALLY PROPELLED VESSEL.

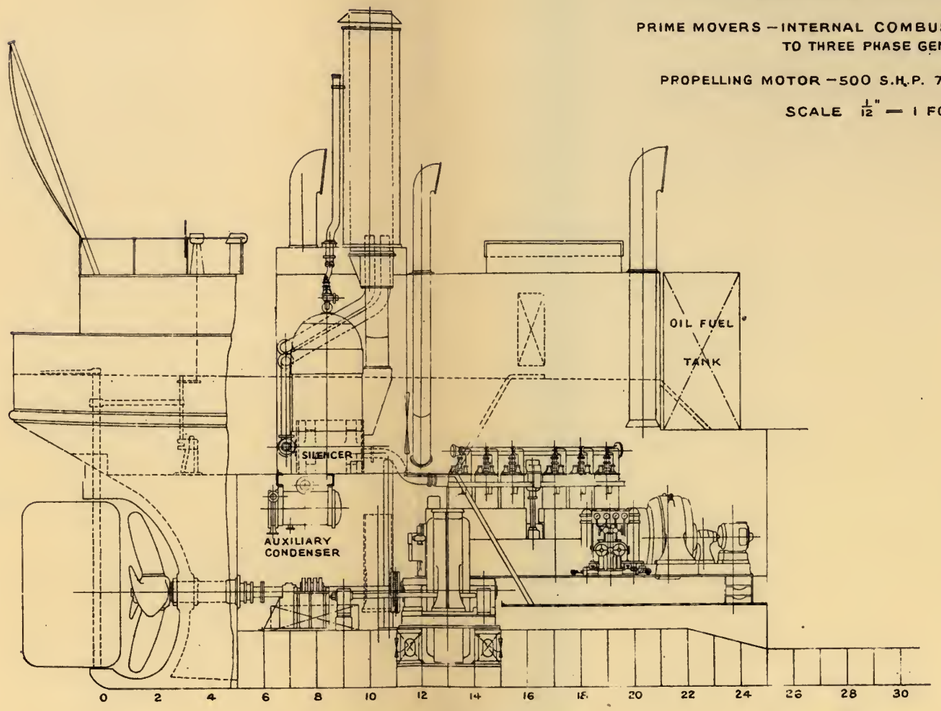
(MAVOR SYSTEM)

FOR SERVICE ON THE CANADIAN LAKES & CANALS.

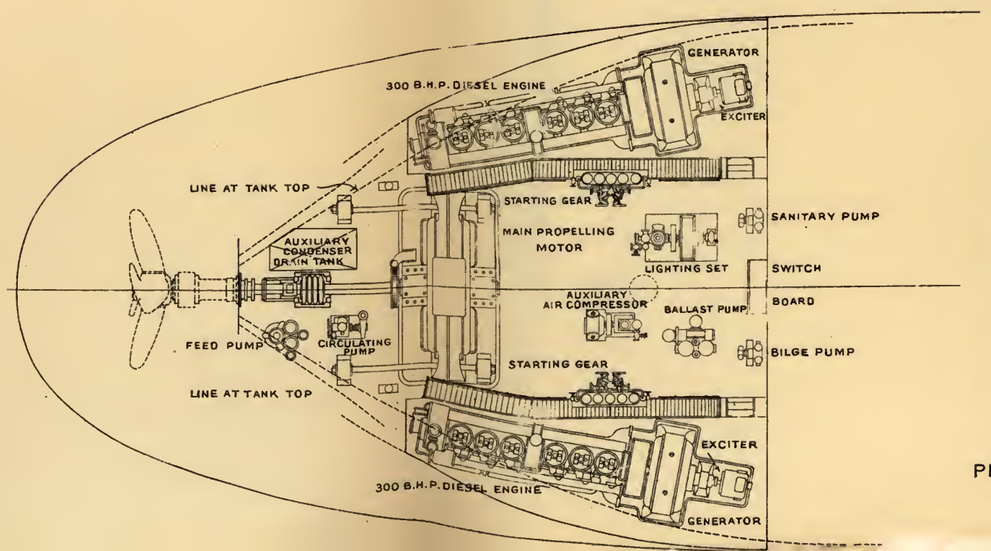
PRIME MOVERS — INTERNAL COMBUSTION ENGINES COUPLED TO THREE PHASE GENERATORS 400 REVS. PER MINUTE.

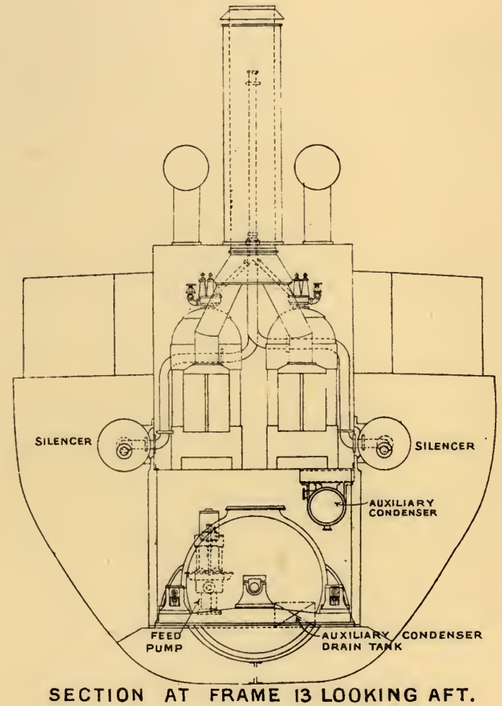
PROPELLING MOTOR — 500 S.H.P. 78 REVS. PER MINUTE.

SCALE $\frac{1}{12}$" — 1 FOOT.

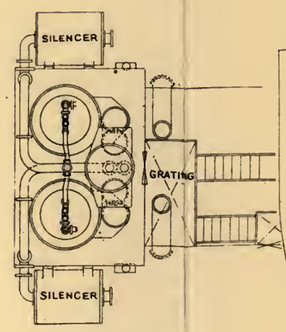


SECTIONAL ELEVATION.

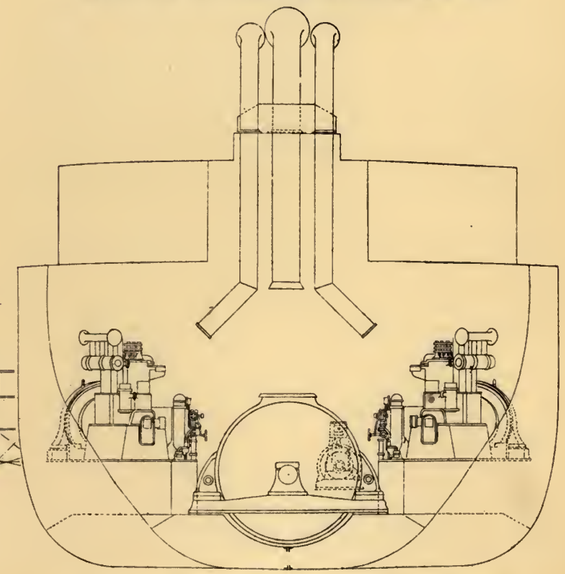




SECTION AT FRAME 13 LOOKING AFT.



PLAN OF BOILERS.



SECTION AT FRAME 13 LOOKING FORWARD.

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3 0112 018312824